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THE DESIGN OF AN EXPERIMENTAL FURROW OPENER UNIT FOR DRY LAND SEEDING OPERATION

by

Brundaban Tripathy ©

A THESIS

SUBMITTED TO THE FACULTY OF GRADUATE STUDIES AND RESEARCH

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The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research, for acceptance, a thesis entitled THE DESIGN OF AN EXPERIMENTAL FURROW OPENER UNIT FOR DRY LAND SEEDING OPERATION submitted by BRUNDABAN TRIPATHY in partial fulfillment of the requirements for the degree of Master of Science in Agricultural Engineering.

ABSTRACT

Approximately 40 percent of the Indian agricultural production is from dryland areas which occupy about 75 per cent of the total arable land in India. The availability of moisture is acute during the rabi (winter) season. Crop establishment is difficult due to dry surface soil conditions often to a depth of 4.0 to 6.0 inches. The germination of the seeds is greatly improved by forming furrows to push aside the dry soil and planting in the bottom of the furrow where there is moisture.

A heavy duty hoe drill or soil lifting opener of similar kind, which can open furrows up to a depth of 6.0 inches, has the critical problem of high energy requirement coupled with variations in furrow depth. The latter problem gives rise to irregular emergence of seedlings that ultimately results in wide differences in crop yield within a row as well as between the adjacent rows. On the other hand, a disk opener that goes to a depth of 6.0 inches makes the furrow width too large.

A triple disk unit followed by a rotor was mechanically designed to be used as an experimental furrow-opener device in dryland agriculture in India. The disks open a furrow of 4.5 inch depth, removing the top dry soil. The rotor forms a 'U' shaped furrow of 1.5 inch depth beyond the depth of the furrow opened by the disks. The unit was tested in silt loam soil of of the Ellerslie farm of the University of Alberta. Percentage

of seedling emergence of wheat and rapeseed were taken into account for determining the performance of the soil-opener device and selecting the optimum design parameters i.e., the width of the rotor blade and the height at which the two disks in the double disks unite, at the front. The energy requirement of the opener was determined.

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CHAPTER I

INTRODUCTION

Dry land agriculture is of considerable importance in India. About 75 per cent of India's total cultivated land is rain fed (21). India is a monsoonal country, the maximum rainfall generally comes during its monsoon season, preceded by a dry summer and followed by a comparatively drier winter. This phenomenon is particularly observed in states like Madhya Pradesh, Maharastra, Rajasthan, Haryana, Andhra Pradesh and Tamilnadu (12). Soil in most of these states is alluvial or sandy loam in texture (21) and light in density. The average annual rainfall is 15 inches and ranges approximately from 8 to 25 inches. These states, therefore, grow khariff crops (summer crops) between the last week of June and the last week of September. The rabi crops (winter crops) are grown between October and May, with the moisture left in the soil after the harvest of the previous khariff crops(12). The problem associated with the raising of rabi crops is that by the time the seedling of the rabi crop is to start, the surface soil becomes dry to a depth of 3 to 6 inches depending upon the moisture stored from the khariff season(12). The hydrograph (Fig. 1) shows the rapid fall of the water table between August and October. Although rainfall is expected during the last week of October, it is very much variable and unpredictable, and inviting the problem of late seeding which

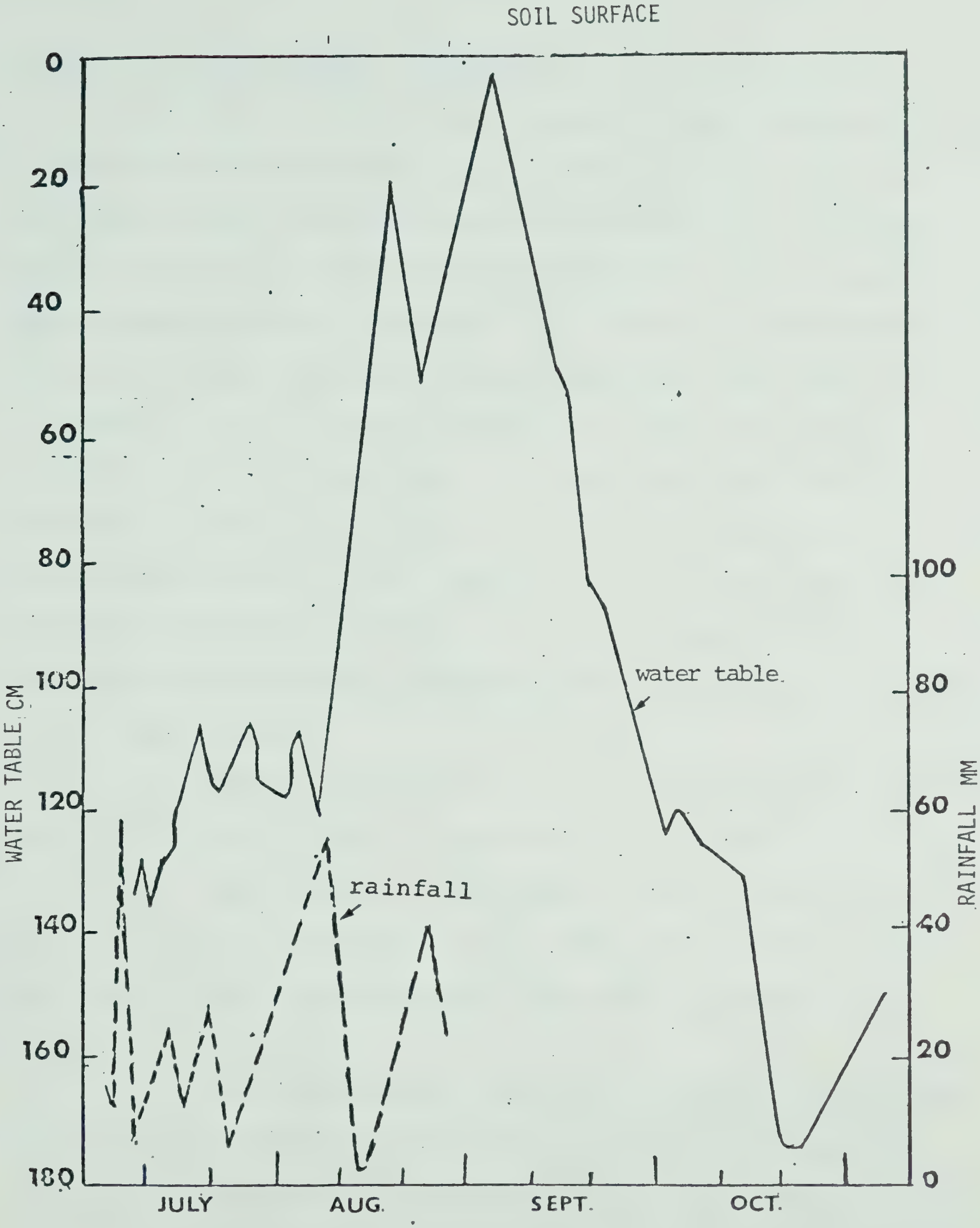


Figure 1. Hydrograph (Typical) (Leeuwrik) (24).

results in a considerable decrease in grain yield.

A solution to this vital problem of rabi cropping is the development of a seed drill that will remove the top dry soil and will place the seed in the moist soil, simultaneously providing an ideal seed environment for uniform and rapid plant emergence. In fact, crop yields are influenced not only by final stands but by the uniformity and rapidity of emergence (22). In this context the seed furrow opener or forming device is definitely the most vital component of the planter, since configuration of the furrow formed by the opener determines the final position of seed in the soil (22). Under field conditions there are several principal factors that define the operating characteristics of the furrow opener.

These are:

1. The degree of precision which can be exercised over depth of penetration in the soil by the opener.

2. The ability of the furrow opener to form a clearly defined uniform seed furrow under a wide range of soil types and moisture conditions and wide range of depth of operation in the soil.

3. The trash control and cutting ability of the opener blade, when the opener is used for direct drilling operation.

4. Positive seed insertion with firm seed-soil contact.

5. Wear of soil working elements not interfering with seed deposition or increased load requirement.

The conventional furrow openers generally consist of:

1. Fixed types
2. Rotary types

Hoe, stub runner and curved runner come under the category of fixed types. Double disks, single disks, and double disks with a coulter at the front come under the category of rotating types. The rotating types may be powered to rotate (2) or simply rotate by friction with soil. The runners are suitable when the desired depth of seed placement is from shallow to medium, but unsuitable for conditions, where the depth of seed placement can be extended up to 6 inches. Runners are unsuitable due to the fact that the moist soil is overlaid by dry tough soil which must be pushed aside which results in high friction and draft requirement. Furthermore when these types of openers are used in the above soil conditions, the volume of soil removed varies, thereby causing variations not only in the original depth but in covering depth. The same problem, though to a smaller extent, can be envisaged in the case of disks which rotate by friction with the soil. In general, any soil opener which depends upon wedging soil aside and is expected to go to a greater depth will produce variable openings due to changes in amount of friction caused by soil texture and moisture (11). On the other hand, disks which are rotary powered require higher power for their operation and cause excessive soil pulverisation which is undesirable for a good root bed (15). Moreover, the

single disk furrow opener cannot be used for the specified typical Indian dry land seeding operation. The reasons are (1) the dry soil will not be pushed aside from the top, (2) a clean furrow will not be formed, and (3) the height of soil over the seed will be more than 2.0 inches. This in turn will necessitate higher energy requirement on the part of the seedling to emerge and ultimately the germinated seedling may not be able to come to the top. This problem which is associated with a single disk opener can be overcome using a double disk opener; but the double disk will have a much larger draft at a depth of 6 inches, compared to a single disk opener. Gordon (17) found out that using disks of 26.0 inch and 20.0 inch diameter, the draft 'L', vertical reaction 'V' and side thrust 'S' decreased when disks of larger diameter were used. The soil type in this experiment was Davidson loam, the depth of cut varied from 4.6 inches to 7.0 inches and the tilt angle of the disks were 0°. Hence due to reduced upward thrust, the larger disks tend to penetrate better. A double disk opener with disks of larger diameter and of single concavity, is also unsuitable for dryland seeding operation. Even providing minimum disk angle and zero tilt angle the cross sectional area of the furrow becomes too large, distance between the midpoints of two consecutive furrow sections becomes large and there is poor seed concentration in the midzone. In addition, to open a furrow of 6.0 inch depth, disks of less than 24.0 inch diameter

should not be used (16). The alternative is to open a furrow of 4.5 inch depth using disks of 16.0 to 18.0 inch diameter and to open an additional furrow of 1.5 inch depth beyond 4.5 inches by some other furrow opening device. Disks with less than 16 inches in diameter should not be used, because the smaller the diameter the larger is the draft requirement and the poorer is the penetration. The general problem associated with the disk openers is that the V shaped furrow section does not provide optimum seed soil contact when the shape of the seed is irregular or the size of the seed is large.

Therefore the objectives of this project were:

1. To design the main parts of a furrow opener device and to fabricate the opener unit that will open up a furrow of configuration as shown in Figure 2 and to meet the requirements of an ideal furrow opener as specified earlier.

2. To test the opener so as to select the optimum parameters involved in the design.

For this purpose a furrow opener with the following components was designed (Figure 3 and 4).

1. A smooth running plain coulter of 17 inch diameter, at the front of the opener assembly.

2. A double disk opener, each disk of 18 inch diameter and 23.3 inch pitch circle radius with a disk angle of 23^0, running behind the coulter and capable of opening a furrow to a depth range of 2 inches to 4.5 inches depending upon the thickness of dry soil. The double disk opener was provided with an adjustable depth control unit.

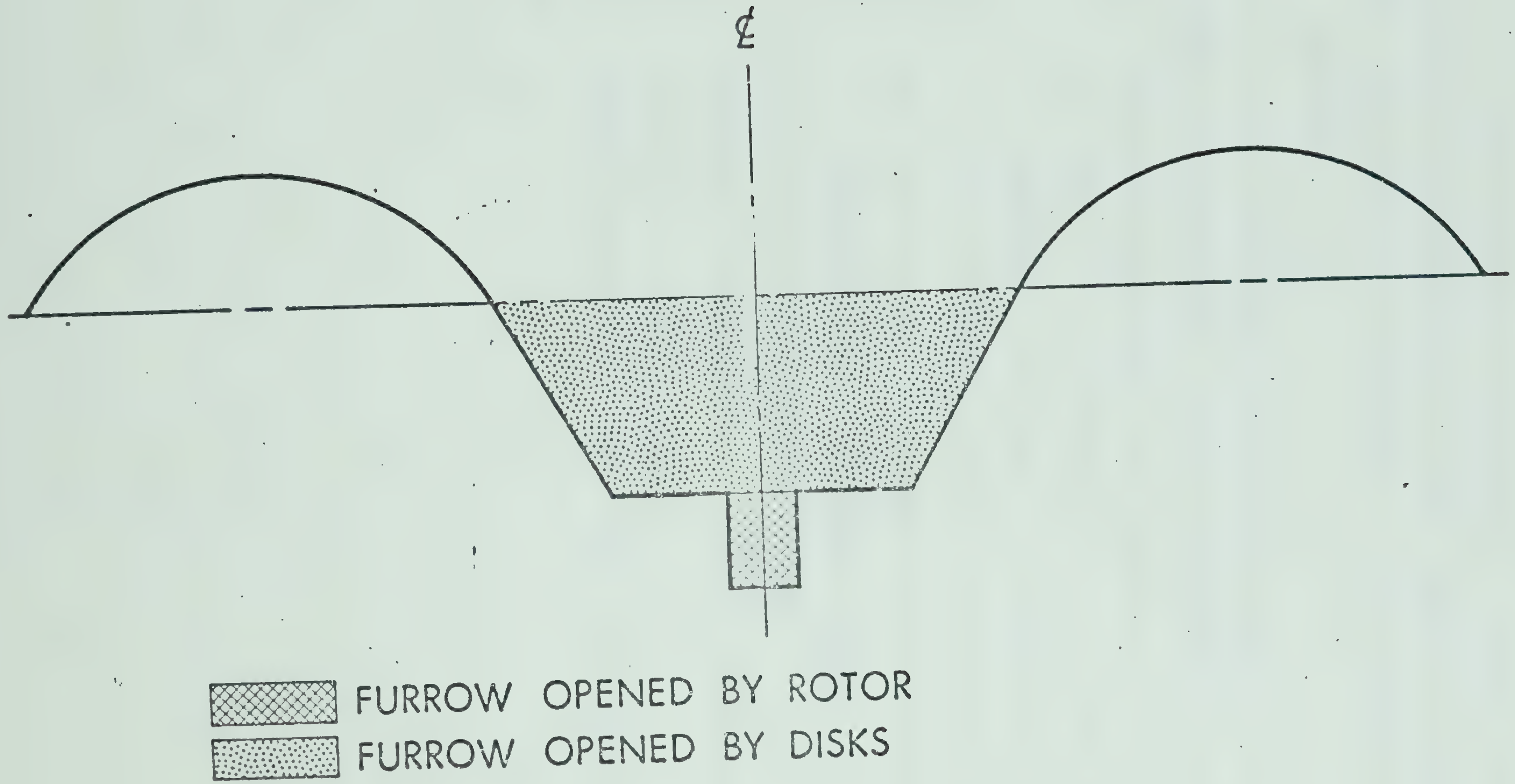


Figure 2. An Ideal Furrow Configuration.

3. A rotor capable of opening a U shaped furrow of 1.5 inch depth, below the depth of soil opened by the double disk unit. The rotation is opposite to the direction of movement of disks.

4. A seed spout that is extended directly into the furrow behind the theoretical circle described by the rotor.

5. The covering device which consists of:

(a) Two covering blades set wide at the front and narrow at the back in the horizontal direction depositing the moist soil into the trench.

(b) An inverted 'V' shaped packing wheel running behind the soil covering blades.

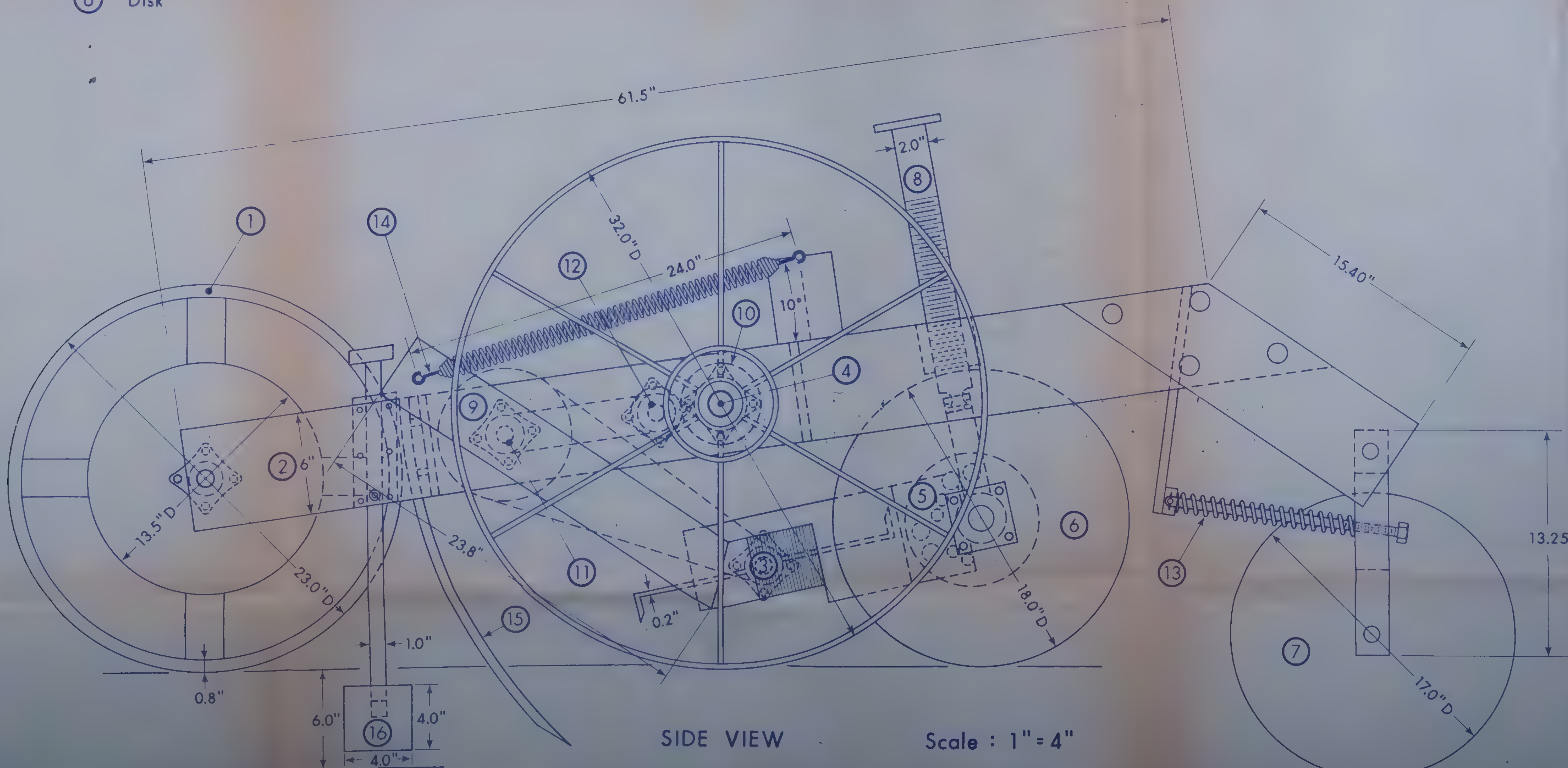
The selected design parameters were:

1. Coulter, its presence or absence in the machine.

2. Width of the rotor blade.

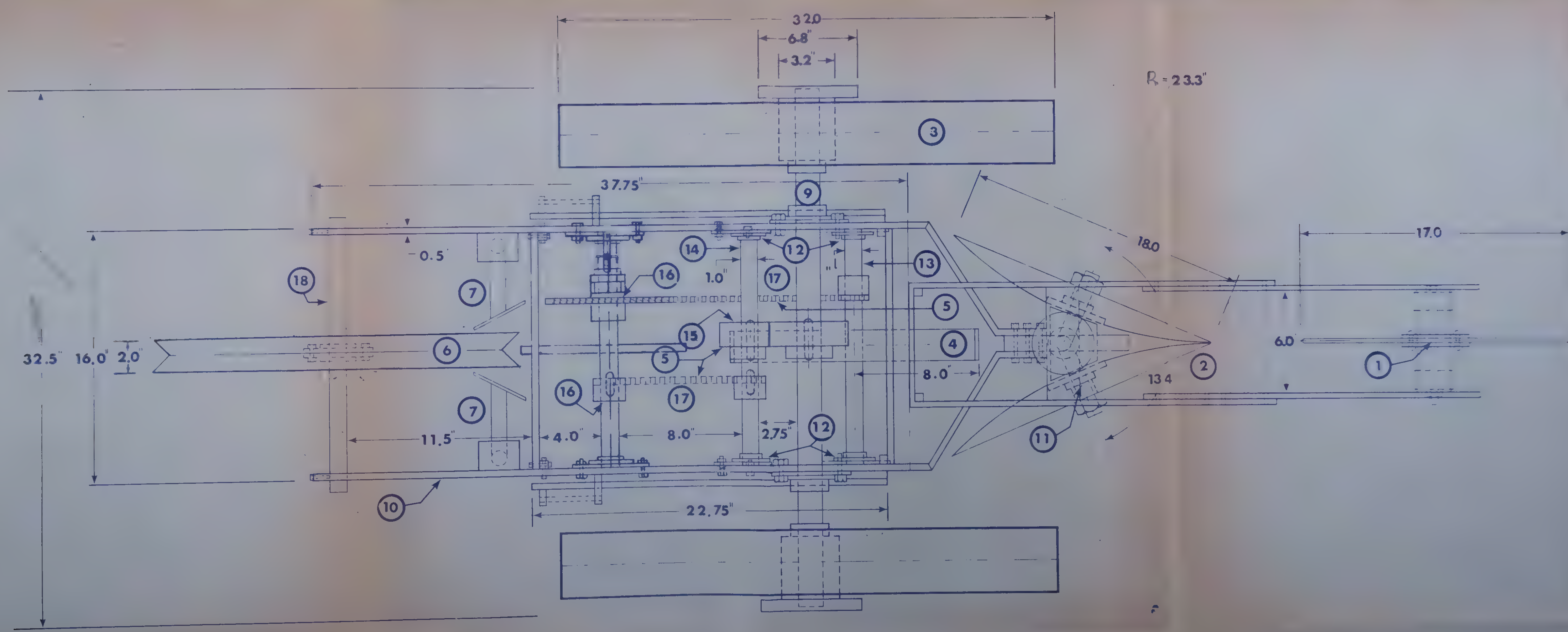
3. Height at which the two disks unite in a double disk unit.

(1) Soil Press Wheel
(2) Frame
(3) Rotor
(4) Main Axle
(5) Double Disk Housing Unit
(6) Disk
(7) Coulter
(8) Depth Control Screw
(9) Sprocket
(10) Gear
(11) Sprocket
(12) Sprocket
(13) Comprssion Spring
(14) Tension Spring
(15) Seed Spout
(16) Soil Covering Blades



EXPERIMENTAL FURROW - OPENER UNIT. Fig. 3

1. Coulter and Housing
2. Double Disk Unit
3. Walking Wheel
4. Rotor
5. Speed Multiplication Unit for the Rotor
6. Press Wheel
7. Soil Covering Blades
8. Seed Spout
9. Main Axle
10. Frame of the Machine
11. Deep Grove Ball Bearing
12. Adapter Bearing
13. Rotor - Axle
14. Auxiliary Axle
15. Spur Gear and Pinion
16. Sprocket
17. Chain
18. Press - Wheel Axle



TOP VIEW

EXPERIMENTAL FURROW - OPENER UNIT   Scale: 1" = 4"   Fig. 4.

CHAPTER II

REVIEW OF LITERATURE

The review of literature that follows, relates to the research conducted on rotary tillers and furrow opener development and is divided into three parts.

1. Definition of the scientific terms frequently used.

2. History of furrow opener development.

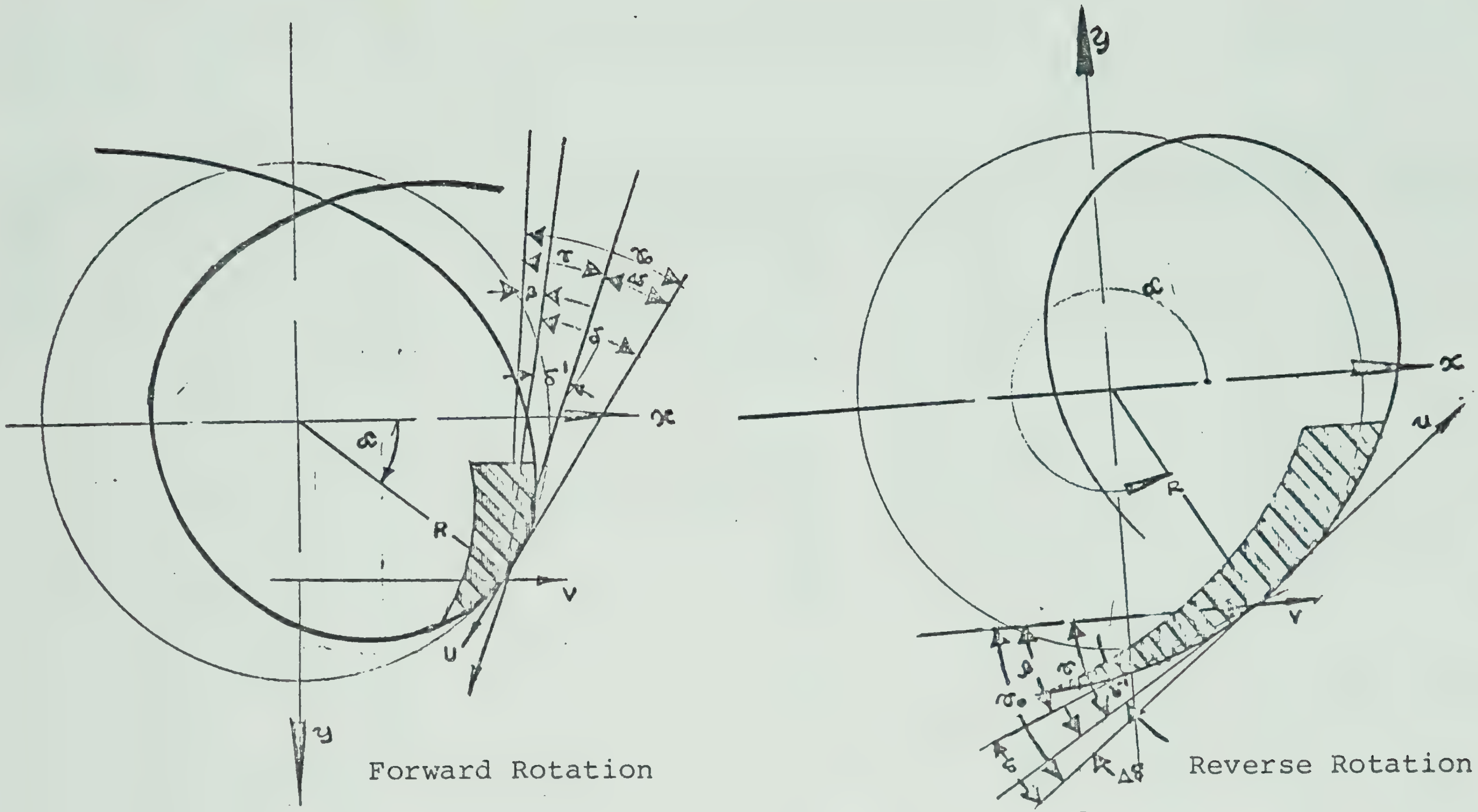
3. Literature review on rotary tillers.

2.1 Definitions

Apparent clearance angle. The apparent clearance angle 'δ' shown in Figure 5 is the angle between the back surface of the sharpened edge and the tangent to the rotor circumference (19).

Effective cutting angle. The effective cutting angle 'γ' shown in Figure 5 is the angle between the front surface of the blade and the trochoid (surface of undisturbed soil) (19).

Effective clearance angle. Effective clearance angle 'δ' shown in Figure 5 is the angle between the back surface of the blade (sharpened surface) and the trochoid (19).



Forward Rotation

Reverse Rotation

Figure 5. Angle of cutting with forward and reverse rotation.

α, angle of rotation;

β, sharpening angle of blade'

γ, effective angle of cutting;

γ_0, apparent angle of cutting;

δ, apparent clearance angle;

$\Delta\delta$, intersection angle;

δ^1, effective clearance angle (Bernacki)(4).

Disk angle. Disk angle 'α' shown in Figure 6 is defined as the angle, viewed from the top of the disk facing the direction of travel, between the plane of the disk edge and the direction of travel (2).

Draft. Draft is defined as the horizontal component of pull in the direction of motion (2).

Intersection angle. Intersection angle. Δδ' shown in Figure 5 is the angle between the tangent to the rotor circumference and the trochoid at the point of intersect (19).

Specific soil resistance. The specific soil resistance 'K' is defined as the ratio of normal force 'N' on the wedge of the cutter and the area 'F' of the wedge of the cutter (16).

Tilt angle. The tilt angle 'θ' shown in Figure 7 is defined as the angle viewed from the side of the disk, between the plane of the disk edge and vertical line (2).

2.2 History of Furrow Opener Development

The existing furrow openers are capable of opening furrows from medium to shallow depths. There are some hoe and shovel types which with the addition of extra weight may be used for deep placement of seed (2), but additional problems consisting of high draft requirement and large furrow width would be involved.

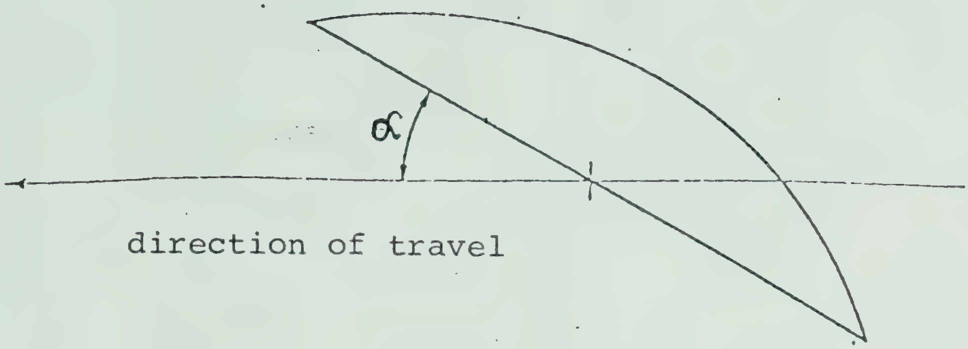


Figure 6. Disk angle.

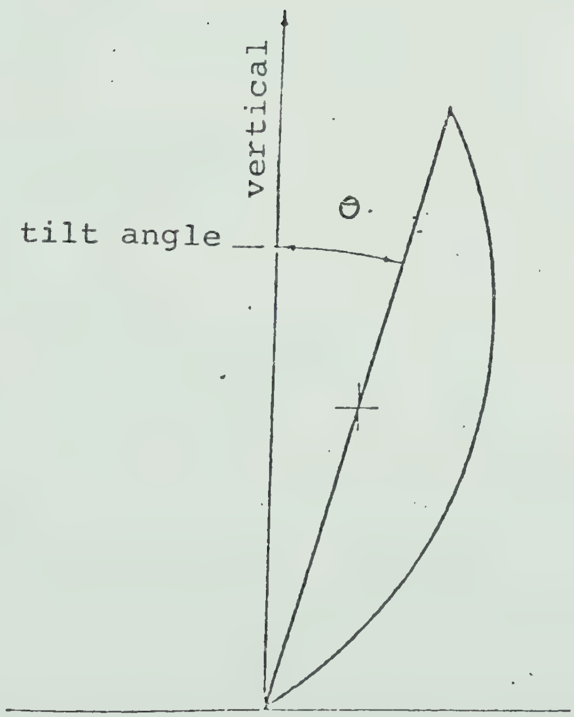


Figure 7. Tilt angle.

Modification of the existing furrow openers for dry land seeding operation would therefore be necessary. The modified furrow opener units should go to a depth of 6 inches and should solve the aforesaid problems. Hence, it is essential at this stage to be familiar with the types of openers associated with conventional drilling operation. The two main types are:

1. Fixed types.
2. Rotating types.

2.2.1 Fixed Types

The position of these openers stays fixed relative to the seed drill. They consist of (a) runners, (b) shovels.

Runner Openers

(i) Curved Runner: The opener has a curved blade and no moving parts. It is usually used with cotton and corn seed-drills or planters (2).

(ii) Fully Curved Runner: The fully curved runner as shown in Firgure 8, is similar to a curved runner excepting that the tip of the former is more curved than that of the latter. The upward curved tip rides over the trash, thus planting the seed rather close to the surface (33).

(iii) Stub Runner: The stub runner shown in Figure 9 has a sharper point than the other two curved runners. Hence, it plows through the trash rather than riding over the trash.

Figure 8. Fully curved runner.

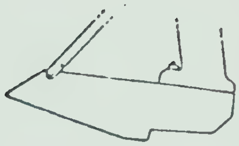


Figure 9. Stub runner opener.

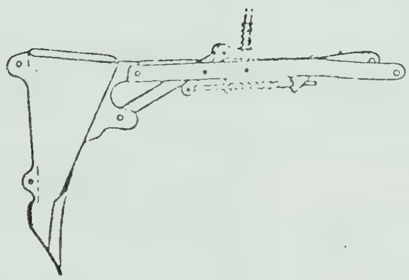


Figure 10. Hoe opener.

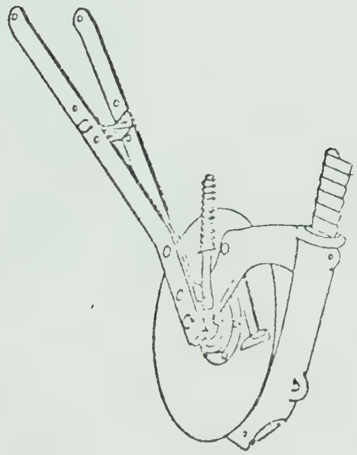


Figure 11. Single disk opener.

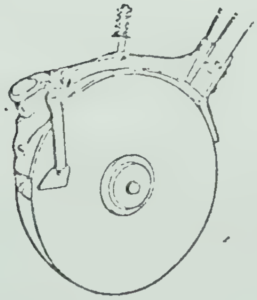


Figure 12. Double disk opener

This type is best for a rough bed or for one filled with quack grass (33).

Shovel Opener or Hoe Opener

The hoe opener shown in Figure 10 consists of a single or double pointed hoe, fastened to a solid cultivating standard. A boot is welded to the rear of the standard to receive the spout end of the seed tube. A spring or pin trip is usually provided so that when the hoe strikes an obstruction, no damage is done to the opener blade. The disadvantage associated with the opener is that it gives trouble by clogging up when used in trashy ground (2). Clogging may occur due to the geometry of the individual opener or the arrangement of openers in a drill.

2.2.2 Rotating Types

They consist of disk openers which are of different types:

(i) Single Disk Openers: The single disk opener shown in Figure 11 consists of one curved disk slightly dished, securely fastened to the boot and set up to run at a small angle. The seeds are dropped from the boot on the convex side at a point below and to the rear of the centre. A toe scraper is used on the convex side and a 'T' scraper on the concave side to keep the disk clean. This type of opener with adequate diameter gives good penetration, cuts trash well and does not easily clog (2).

(ii) Double Disk Opener: This type of opener shown

in Figure 12 is composed of two disks, having very little dish (curvature) or none at all, set facing each other at a slight angle so as to form a bevel cutting edge where they penetrate in the soil. In this position the disks open a clean furrow and leave a small ridge in the centre, so that when the seeds are deposited in the furrow, there is a tendency to make a distinct row.

2.2.3 COmbination of Rotating and Fixed Types

In addition to the above types of furrow openers, there have been combinations of (1) disk openers and runner or hoe openers, (2) disk openers with additional attachments, (3) triple disk openers. Based on different combinations, numerous research projects have been carried out in different parts of the world. The projects are discussed below.

Lister Opener

The lister opener uses a disk but is provided with additional attachments. The opener can form deep trenches and ridges in the soil so that snow and moisture can be caught and the soil can be prevented from blowing by wind. The spacing of the openers is wider than that used for the regular grain drill and ranges from 12 inches to 16 inches between openers (33).

In 1972, at Plant Protection Limited, Fernherst, England, and experiment (22) was carried out using a special test ring with provision for using (1) a knife and disk opener,

and (2) a double disk opener with coulter in the front i.e., triple disk opener.

The knife and disk opener had a knife blade at an angle to a plain disk or coulter running in the front.

The triple disk opener consisted of a double disk arrangement in which the disks had no dish at all and there was a flat coulter in the front of the double disk arrangement.

A comparison was made between the above two units. The results are shown through Figures 13a, b, and c. The analysis showed that the triple disk system was nearer to specification than the other unit. For example, the load required to hold the coulter at a desired depth increased in direct proportions to depth for both the units, but the triple disk showed less load requirement above 2.5 inches to 3 inches both for the tractor pull and for the coulter loading. For any tractor pull, the coulter load requirement of the triple disk, was far less than that of the knife and disk , at all depths. The triple disk also possessed a further advantage in that a smooth running coulter worked better in trashy conditions.

John Deere has developed a rotary disk lister (26) with hard ground opener which is a three way combination.

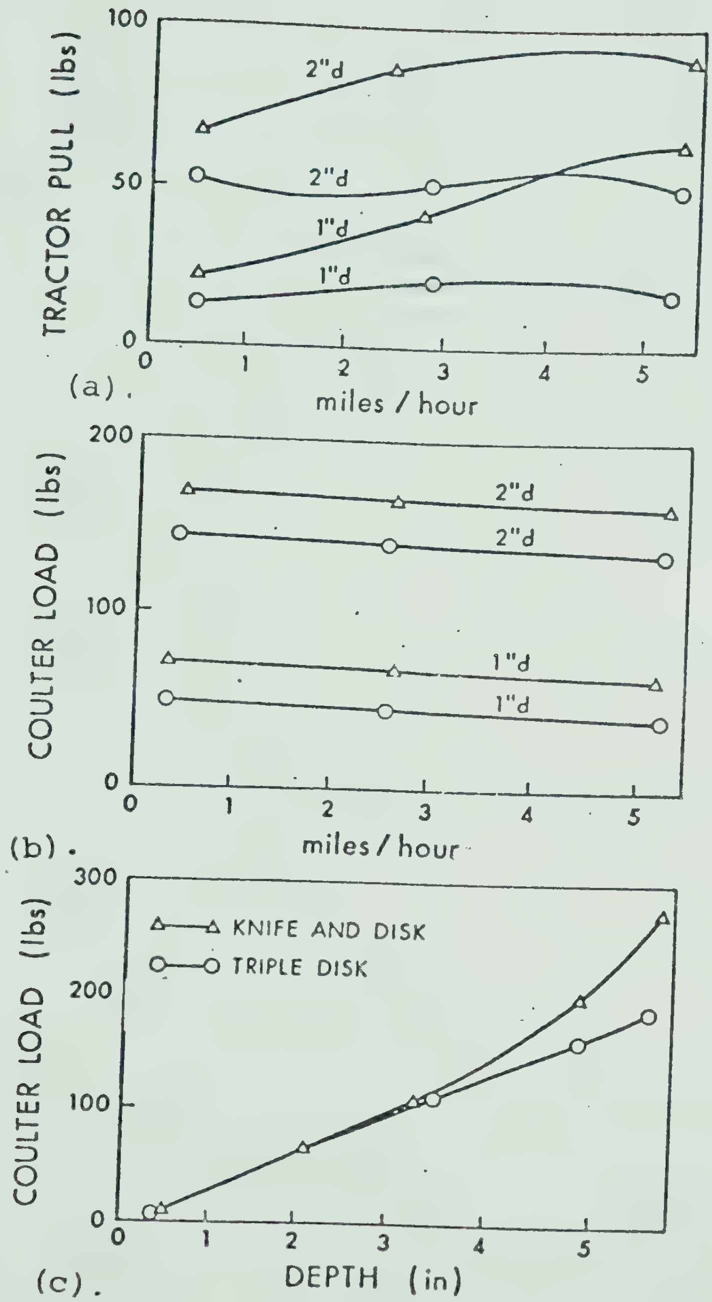


Figure 13. Comparison between Triple Disk, and Knife and Disk Openers.
d = depth of furrow.

1. A shovel type hard ground opener running at the front.

2. Two curved opener disks following the shovel.

3. A seed furrow opener in between the disks, a little behind the hypothetical line joining the centres of the disks.

The covering device consisted of two curved disks with wider spacing at the front and narrower spacing behind.

Smith et al. (30), showed that a power driven oscillating tool consisting of a disk mounted at an angle on a rotating shaft opens a clean uniform furrow in sod with only moderate power requirement. The action of this tool was such that the variations in soil conditions would have little effect. Using this principle, Futral and Verma (15) recently designed a powered furrow opener for precise depths. A schematic diagram of the opener is shown in Figure 14a. The opener consists of 3 plain steel disks each of 0.25 inch in thickness, 5 inches in diameter and the disks are fastened to a shaft. Six teeth made from 0.25 inch x 1 inch hot rolled steel and milled to a wedge are bolted to each disk. The centre disk is mounted at an angle of 86^0 34' with respect to the shaft and the teeth on this disk extend 1.25 inches beyond its circumference. The outer disks are mounted at an angle of 78^0 54' with the teeth extending 0.75 inch beyond their circumferences. When the shaft rotates a T shaped furrow is formed

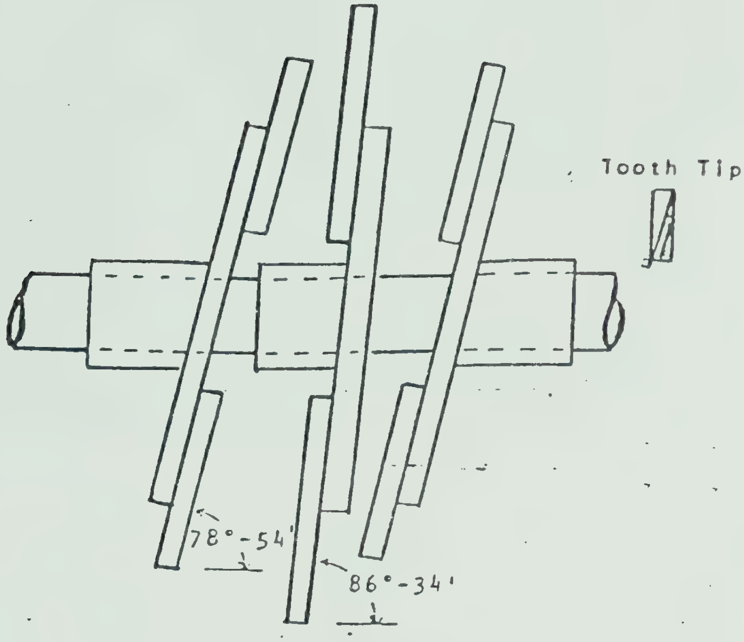


Figure 14a. Oscillating opener for 'T' furrow.

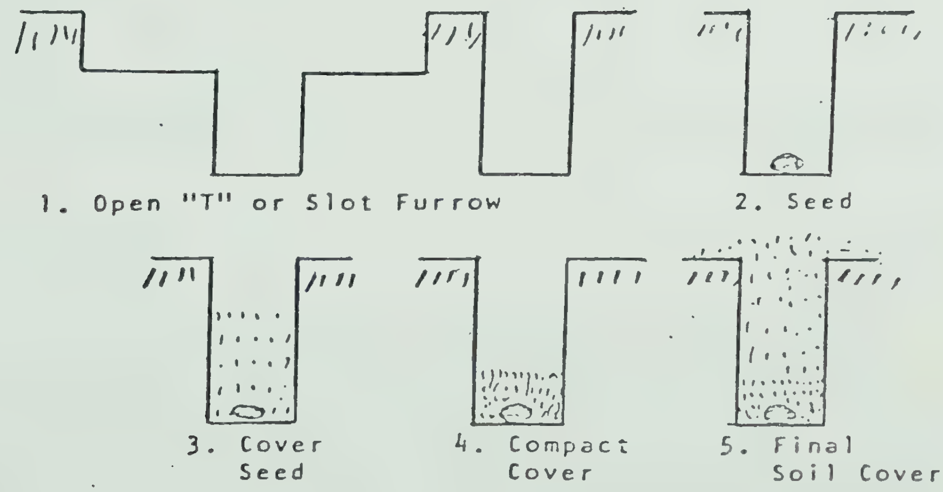


Figure 14b. Sequence of Planting operations.

as shown in Figure 14b. The top of T is 4 inch wide, 0.5 inch deep; the vertical body of the T is 0.75 inch wide with 1.25 inch deep below the original soil surface. Just in front of the opener two rubber tired wheels are mounted for depth control. Preliminary test results showed that no difficulty was experienced in maintaining the depth of furrows on compact surfaces, but the two wheels with 3 inch tires sank into loose soil producing very erratic depth control. Therefore a roller of 14 inch diameter and 18 inch width was put in front of the opener in place of the disks and this worked satisfactorily on loose soil. Final test results revealed that the oscillating rotary seed furrow opener produced superior results compared to a conventional double disk unit planter. The power requirements for this opener were within reasonable limits. The ability of this type of unit (with the use of the disks of larger diameter), to open a furrow of 6 inch depth under the specified Indian soil conditions has not been studied.

In 1973 Dyck and Kataria (12) developed a dryland ridger seeder at the Agricultural Research Station, Swift Current, Canada, which is claimed to be suitable both for khariff (summer) and rabi (winter) cropping at Haryana. In this machine three Acme potato hillers (ridgers) form the ridging assembly. The wings of the potato hillers are adjustable to facilitate the formation of ridges and furrows of various demensions. The shape of the furrow formed can be varied by

changing the angle at which the ridger makes contact with soil. When the angle of approach to the soil surface is small, a nicely rounded furrow with a flat bottom is formed. When the angle of approach is increased a V shaped furrow is formed. At the bottom of this large V shaped furrow a smaller furrow is formed by the nose of the ridger. The hoe type furrow openers used for seed placement, are behind the fertilizer placement device. The depth of seeding at the bottom of the furrow can be adjusted separately from the depth of furrows and ridges by adjusting the position of the hoe opener relative to the tool bore and ridger assembly. The depth of furrows and ridges is regulated by adjusting the gauge wheel at each end. The seeder was tested on clayloam soil of the south farm at the Research Station, when the surface soil was dry to a depth of 4 inches. Winter wheat was seeded at three furrow depths of 3 inches, 4 inches and 5 inches in the last week of July. The seed was placed 1 inch below the furrow depth and the fertilizer 1 inch below the seed. Germination percentages after one week of seeding for winter wheat were 55.2 per cent, 52.6 percent and 50 percent for the furrow depths of 3 inches, 4 inches and 5 inches respectively. This no doubt is a poor result as stated by the researchers. No test was conducted on the draft requirement of the ridger opener. As far as the performance is concerned no comparison was made between this ridger opener and any other conventional opener unit.

2.3 Literature on Rotary Tillers

So far, no work has been done, where a rotary tiller follows the triple disk opener, for the purpose of opening a furrow of greater depth. Hendrick and Gill (19) examined the effects of three design parameters of rotary tillers from the research results of a number of sources. These three design parameters are (1) direction of rotation, (2) depth of operation and (3) ratio of rotor peripheral and machine forward velocities. These parameters have a marked influence on all phases of tiller operations from the power required to the final soil conditions.

2.3.1 Direction of Rotation

According to Hendrick and Gill (19) reverse rotation causes the tiller blade to operate towards an unconfined area (the soil surface) and more of the soil is expected to fail in tension than in the case of forward rotation. The researchers vary in their views about this logic. The results of previous researchers, as cited by Grinchuk and Matyashin (18), showed in general, reverse rotation decreased the force of cutting by 1.5 times, gave better depth stability and reduced breakdown of tools in stony soils. Among the disadvantages associated with reverse rotation were a greater energy requirement when the depth of operation 'h' was less than the rotor radius 'R' and the need to increase the rotary velocity to prevent throwing soil ahead of the tiller.

In a second study Grinchuk and Matyashin (18) reported the results of N.B. Bok, who found that reverse rotation required

12 to 16 percent less power than forward rotation (no mention of draft force). (Here a depth of operation was less than 0.8 R.)

Matsuo (25) found that the power requirements were less for reverse rotation with the same pitch of cut, and that the reduction in power became larger as the soil strength decreased.

The test results of Furlong (14) conducted in 1956 are displayed in Figure 15 which presents a graph showing the rotor and drawbar power requirements for both directions of rotation, with h = 0.45R for each blade shape. In all cases, except blade number '2', reverse rotation required more motor power (blade No. 2 is a pick blade; the rest are some form of 'L', 'C', or other conventional blades). According to Hendrick and Gill (19) further tests revealed that Furlong's observations were related to blade shape, rather than to the direction of rotation.

Hendrick and Gill (19) report that when h<R, reverse rotation throws forward a great deal of soil (some of which can be controlled by shield design). Furlong (1956) presents some information on the size of clods resulting from both the rotary directions. Furlong concludes that the clod size is generally larger for reverse rotation. Whether large or small clods are desirable depends on the intended use of soil. According to Hendrick and Gill (19) large clods are more desirable after fall plowing for the control of water and wind erosion.

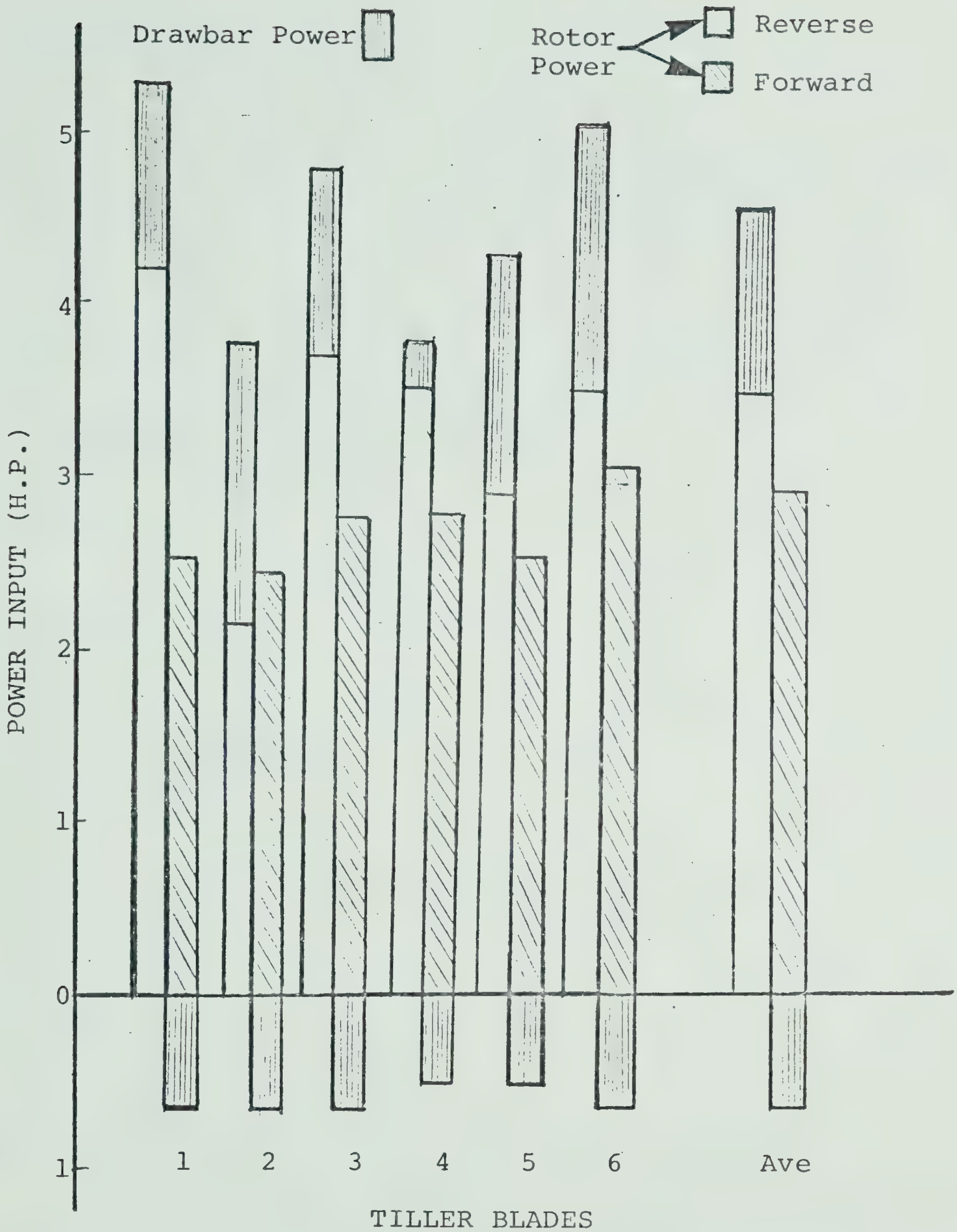


Figure 15. Power Input for Forward and Reverse Rotation.

Blade 1, 3, and 4 are L-shaped blades; 2 is a pick blade; 5 and 6 are 'c' shaped blades. (400 fpm peripheral speed, 4 in. deep, 4 in. length of cut ,24 in. wide.)

(Furlong) (14).

2.3.2 Depth of Tillage

Grinchuk and Matyashin (18) pointed out that there appear to be two schools of thought regarding the relation of rotary tiller radius to the energy requirement. One group contends that, at constant velocity and tilling depth, the specific energy requirement is decreased as the radius of the rotor is increased because the thickness of each slice cut off is decreased. This type of argument was presented during a discussion of Bernacki's (3) characterization of forces.

A second group uses the argument that the expenditure of tilling energy is proportional to the length of cutting path, thus decreasing the tiller radius would result in a decrease in specific power requirement (power per unit volume of soil tilled).

According to Hendrick and Gill (19) increasing tillage depth increases total power requirement, but decreases the specific power requirement in general. The optimum rotor diameter-to-depth ratio appear to be in the range of 1.1 to 1.4, where the minimum specific energy requirement occurs.

2.3.3 Ratio of Peripheral and Forward Velocities

The velocity parameter most frequently used, both as a design and use parameter, is the dimensionless ratio of the rotor peripheral velocity to the machine forward velocity and is given by:

$$\lambda = RW/V$$

Where R = Rotor radius

W = Rotational velocity of rotor

V = Machine forward velocity

There are, therefore, three ways (and combinations thereof) in which λ can be varied: change the rotor radius (R), which is seldom done; change the rotor velocity (W), which is common; and change the machine-forward-velocity (V), which is common. Hendrick and Gill (19) analysed the results of various research workers and reported that: (a) decreasing λ by increasing forward speed results in an increase in the power requirement, but a reduction in specific power (provided the geometry of the soil-tool system is not varied too greatly); (b) decreasing λ by decreasing the rotary velocity decreases the power requirement and specific power (again this occurs with the condition that the geometry of the soil tool system bears a favourable relationship to the power requirement;) (c) At a value of $\lambda = 2.4$, the power required is minimum (irrespective of blade shape).

2.3.4 Blade Shape

Effective angle of cutting for rotor blade. According to Adams and Furlong (1) L shaped blades are better than hook-shaped or pick-type (pointed) blades in trashy conditions and are more effective in killing weeds. The former types of blades do not excessively pulverize the soil which is a desirable feature

in dryland seeding operation where moisture conservation can be effectively made by leaving the trash on the field after harvest of kharif crops and maintaining larger clods during tillage. The effective cutting angle is mathematically expressed (1) as:

$$\gamma = \gamma_0 - \Delta\delta = \beta + \delta - \Delta\delta$$

Where β = sharpening angle of blade

δ = apparent clearance angle

$\Delta\delta$ = intersection angle

Bernacki (4) has given a graphical relation shown in Figure 16 between h/R and intersection angle $\Delta\delta$ for various values of λ. .

The magnitude of $\Delta\delta$ depends upon the value of λ, the angle of rotation and the direction of rotation. The smaller is the value of λ, the smaller is the energy requirement, but the greater is the change in $\Delta\delta$ during one cut, and thus, the greater must be angle δ to prevent compaction of uncut soil.

Bernacki (4) has given a graphical relation in Figure 17 between λ, and γ, which can be used to provide adequate blade clearance at different values of λ.

2.3.5 Summary

In addition to the general statements made by various research workers, regarding the effect of the individual parameters, Hendrick and Gill (19) have made some additional

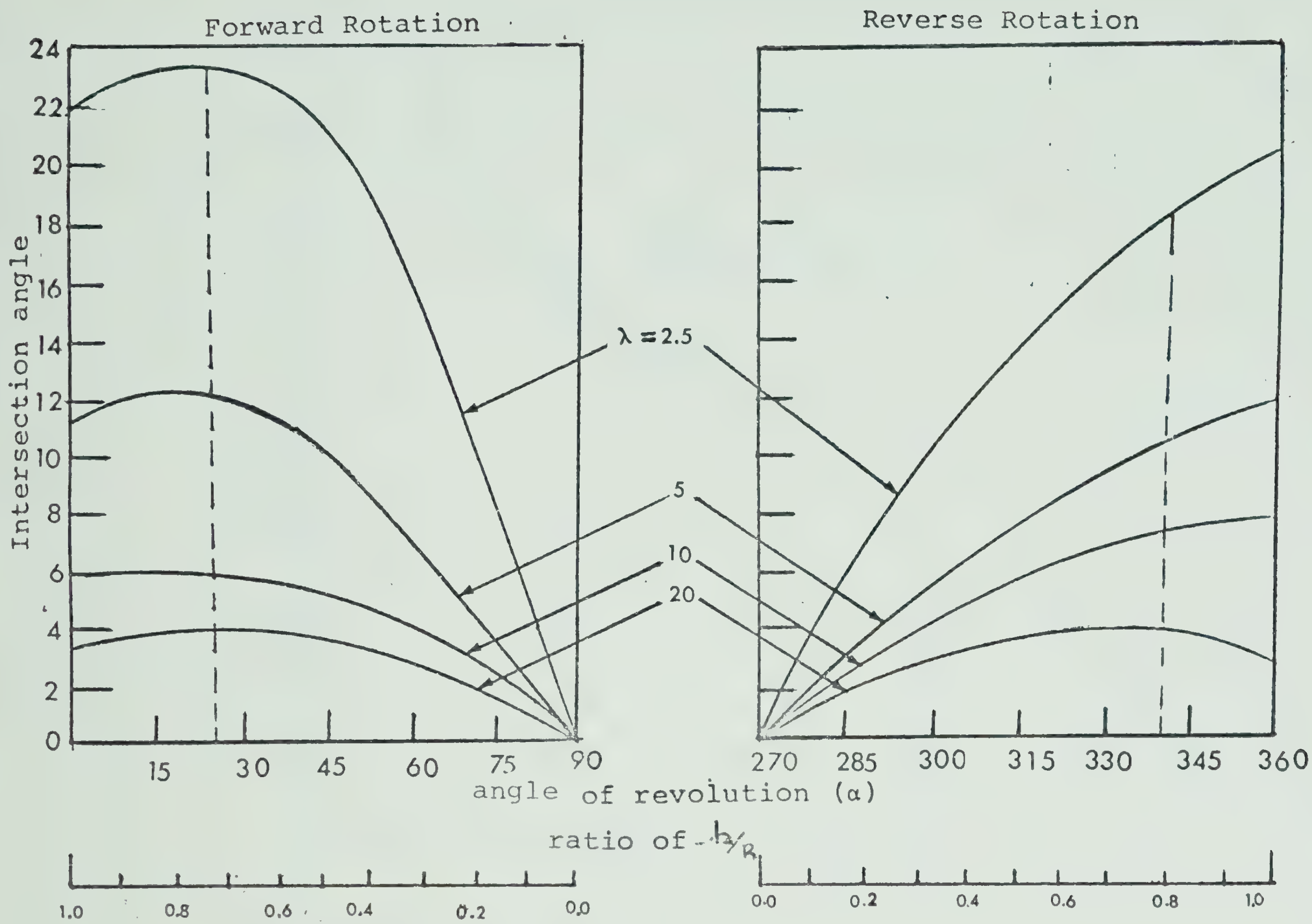


Figure 16. Intersection angle (Δσ) as a function of angle of revolution (α) for various values of λ (Bernacki)(4).

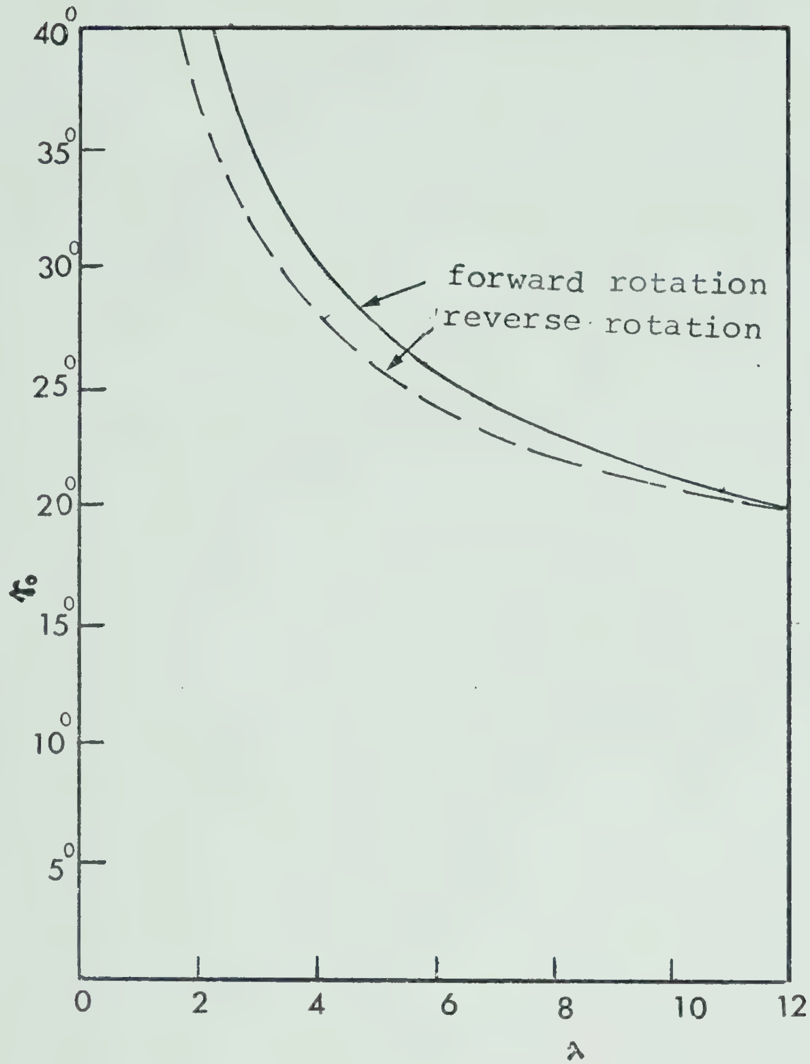


Figure 17. Relationship showing necessary value of γ_o to provide adequate blade clearance at different values of $\lambda = {}^{RW}/_{V}$ (Bernacki) 4).

comments. The most important comment is that while in general some qualitative conclusions can be drawn from the research results of various workers, very little can be gained quantitatively. This is mainly due to the lack of sufficient data and uniformity of presentation. Some authors present descriptions of initial soil conditions, including moisture content of soil and penetration-resistance while others do not. There is no uniformity regarding reporting final soil conditions, description of tiller blades, and occasionally fundamental information about rotor blades is absent.

The prediction of magnitude of power requirement or the performance of rotary tiller from a theoretical basis is not possible at this time. According to Hendrick and Gill (19), no information is available regarding the effects of soil reaction in the range of velocities at which rotary tillers operate. As reported by Hendrick and Gill, other researchers have concluded that the movement of tools through the soil causes considerable variation in the soil reaction and perhaps even in the fundamental behaviour of soil.

Therefore only the fundamental conclusions made by most research workers as reported by Hendrick and Gill (19), and outlined in subsection 2.3.3, can be taken for granted for the purpose of simplifying the present design.

CHAPTER III

MECHANICAL DESIGN

The success of the machine depends upon the accuracy with which the main parts are designed. Hence, the main parts of the machine as reported below were mechanically designed. In all cases economy and commercial availability of the materials were taken into consideration.

MAIN PARTS

1. The disk housing
2. The rotor blade
3. The speed multiplication unit
4. Depth control screw
5. Compression springs and tension springs

The design of the main parts are presented in Appendix 1. The dimensions of all other parts of the machine were selected based on easy commercial availability. In all cases however, the stability and strength of the machine parts and convenience of operation of the machine were taken into consideration.

A pictorial view of the machine is given in Figures 18 and 19.

CHAPTER IV

MATERIALS AND METHODS

4.1 Field and Soil Selection

The soil and the climate to be selected for the purpose of conducting the test had to be similar to the soil and climate for which the experimental furrow opener unit was designed. Consulting the soil survey map of Alberta and the Temperature and Precipitation data of the Prairie Provinces of Canada, over a period of 29 years, the Orthic Brown Soil (Cavendish sandy loam developed on alluvial aolin material) of Bow-Island district was found to be most suitable. Unfortunately in the 2nd week of May, Bow-Island received 4 inches of precipitation. So the experiment was carried out on the Ellerslie farm using the facilities of the Department of Agricultural Engineering, University of Alberta.

4.1.1 Field Size

Two fields with summer fallow condition and of size 200 ft. x 180 ft., were selected for the purpose of testing the performance of the machine. These rectangular fiels were separated by a distance of 30 feet.

4.1.2 Soil Type

Ten soil samples were collected at random from the field, mixed and then analysed in the Soil Science Laboratory of the University of Alberta. The average soil type was

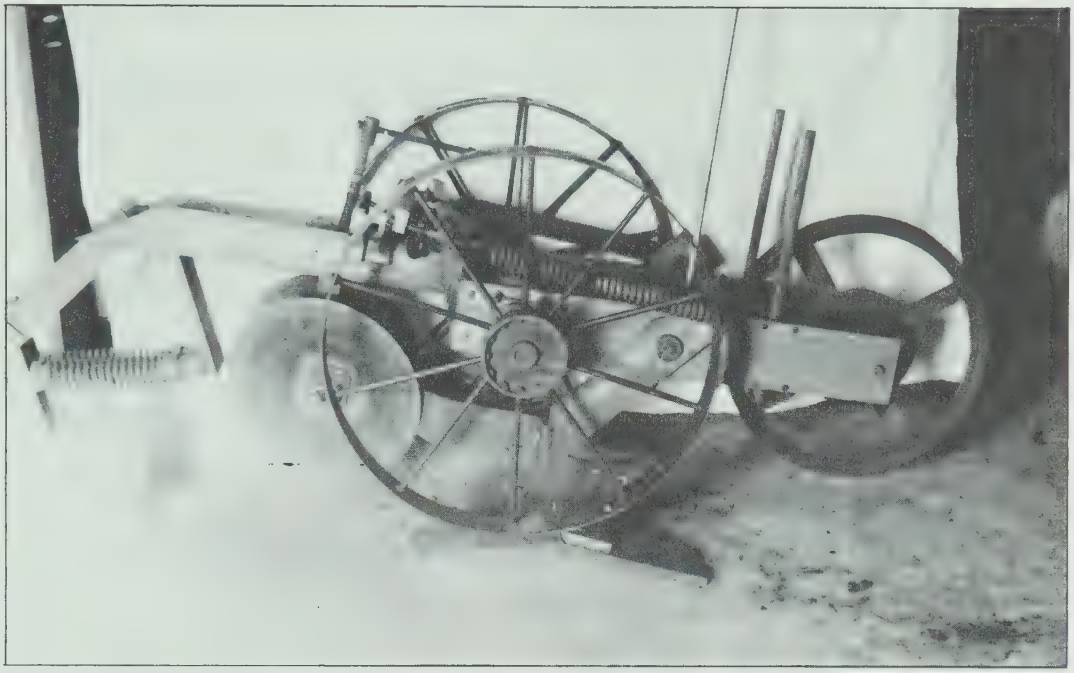


Figure 18. A Pictorial View of the Experimental Furrow Opener Unit

Figure 19. The Experimental Furrow Opener Unit (in the Field)

found to be silt loam in texture and dark brown in colour. The soil had considerable organic matter content.

4.1.3 Soil Moisture and Rainfall

At planting the average soil moisture content from zero to a depth of 4.5 in. from the undisturbed soil surface was 11.5 percent.

The soil moisture content at planting from 4.5 in. to a seed depth of 6 in. from the undistrubed soil surface was 19 percent.

Four days after planting 0.3 in. of rail fell and eight days after planting 0.4 in. of rain fell.

4.2 Field Preparation

Using chisel plows, the fields were cultivated at two day intervals for a period of 3 weeks. The depth of tillage was maintained approximately at 4.5 in. The purpose was to make the top 4.5 in. of soil fairly dry, so as to be suitable for conducting the experiment. A view of the field before the test is shown in Figure 20.

4.3 Equipment for Draft Measurement

The Salter's Improved Dynamometer was used to measure the total pull, needed by the experimental furrow opener unit. The dynamometer is a spring scale equipped with two parallel springs. The scale has twenty divisions and each division is further subdivided into ten. Each sub-division measures a pull of 10 lbs. The dynamometer is

Figure 20. A View of the Field Before Test

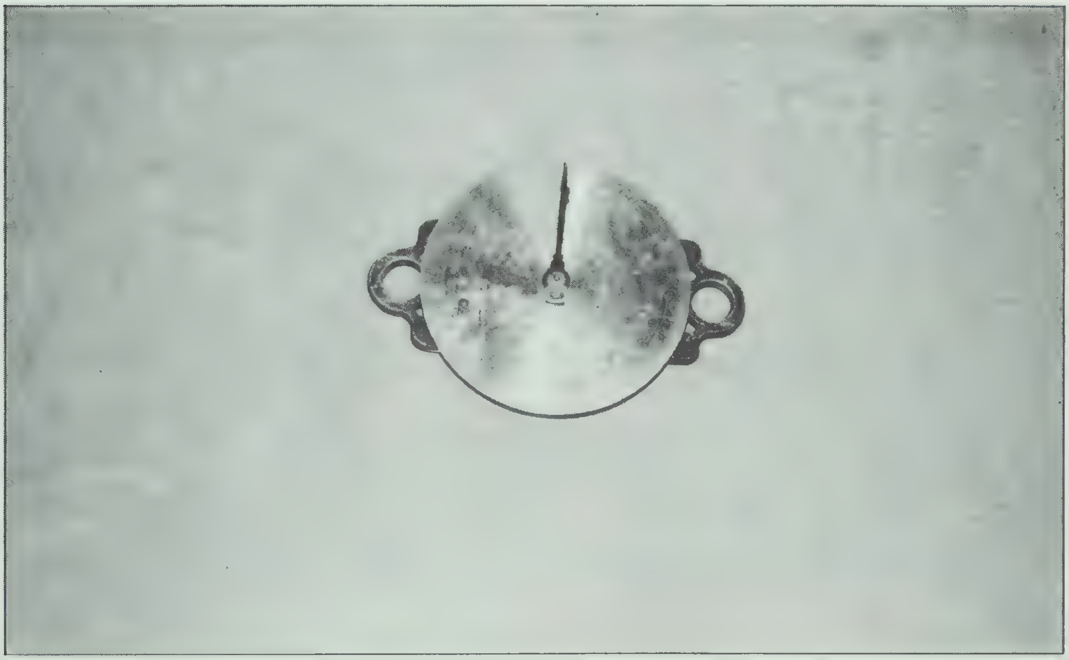


Figure 21. The Salter's Improved Dynamometer

shown in Figure 21.

The scale reading while the machine came across a sudden obstruction was disregarded.

4.4 Design of the Experiment

The parameter that influence the performance of the machine can be divided into two categories

A. Soil Variables

1. Soil moisture

2. Soil density

B. Tool Variables

1. Presence or absence of the coulter

2. Width of rotor blade 0.5 in., 1 in. or 1.5 in.

3. Height at which disks unite (disk height)-4 in., 6.75 in. or 9.0 in.

Soil variables were assumed to be uniform throughout the field, although 25 percent of the field had mulch coverage. The tool variables were taken as independent variables. The pull and the percentage of seeds germinated were the dependent variables. The performance of the machine is measured in terms of the percentage of seedling emergence.

The presence or absence of the coulter in the machine as a tool variable merely increased the scope of experiment and hence its effect on the performance of the machine was determined with least precision. The effects of the other

two variables, height and width were determined with equal precision. The two types of seeds used were wheat and rapeseed, one type of seed being used in each plot.

A split plot factorial design with 3 replications was used in making the tests. The form of analysis is given in Table 4.1 and the experimental order is given in Table 4.2.

4.5 Test Procedure

Each field was divided into three replicates and each replicate into two parts. In order to save labour and for the sake of convenience one part of each replicate was used to test the machine with the coulter attached or coulter removed. Each half replicate was divided into nine subplots. The nine different treatments were randomized following a random number table and each subplot was marked with a peg for the treatment to be applied. A space of 180 ft. x 30 ft. was left in between the adjacent relicates for convenience of tractor movement.

Fifty-four packages of wheat and fifty-four packages of rapeseed, each package containing one hundred counted seeds were used for the experiment. One hundred seeds were planted in a row for each treatment in a subplot of size 10 ft. x 40 ft. where 40 ft. is the length of the row. The seeds were dropped through the seed spout by hand.

The seedlings of the rapeseed began to emerge on the 3rd day. Emergence began during the 5th day after planting

in the case of wheat. Every day in the morning and in the evening two sets of countings were made and this continued for a period of nine days after planting* in case of rapeseed and fourteen days in the case of wheat.

4.6 Method of Analysis

The average number of seedlings emerged on each day was determined from the morning and evening counts. The day on which the emergence rate was maximum for all or maximum number of treatments, was noted for both rapeseed and wheat. The number of seedling emerged on this day and the emergence on the first and last day of the count for each crop were used for analysis. More emphasis was laid on the data of the middle day mentioned above rather than the data of the first and last day. For each set of data best performance of the machine was judged on the basis of maximum percentage of the seedlings emerged. Analysis of variance was used to determine if each parameter and their various interactions had an effect on the seedling emergence.

The pull,** measured for each treatment, during the experiment, formed two sets of data, one for the wheat plot and another for the rapeseed plot. Analysis of variance was also

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*Only fully emerged plants were counted. It was noted that following the rain, the plants that remained in the crook stage for several hours usually died before emerging completely.

** Pull was measured to see if best performance of the machine matches with minimum pull or not.

TABLE 4.1

FORM OF ANALYSIS

Source of Variation	Degree of Freedom
Replicates (R)	2
Coulter (C)	1
Error (1)	2
Sub-total (1)	5
Height (H)	2
Width (W)	2
W x H	4
C x H	2
C x W	2
C x H x W	4
Error (2)	32
Total	53

TABLE 4.2

EXPERIMENTAL ORDER

	Plot Number	Replicate 1 C	WH	Replicate 2 C	WH	Replicate 3 C	WH
	1	1	23	2	12	2	12
	2	1	13	2	23	2	33
	3	1	21	2	33	2	23
	4	1	11	2	32	2	11
	5	1	12	2	13	2	22
	6	1	33	2	11	2	13
	7	1	22	2	21	2	31
	8	1	32	2	22	2	21
Field 1	9	1	31	2	31	2	23
Wheat	10	2	12	1	21	1	23
	11	2	13	1	33	1	11
	12	2	22	1	11	1	22
	13	2	32	1	12	1	32
	14	2	33	1	31	1	31
	15	2	11	1	13	1	13
	16	2	31	1	22	1	31
	17	2	23	1	32	1	33
	18	2	21	1	23	1	12
	1	2	21	2	22	1	32
	2	2	33	2	11	1	21
	3	2	12	2	32	1	31
	4	2	23	2	21	1	13
	5	2	22	2	23	1	12
	6	2	11	2	12	1	23
	7	2	31	2	13	1	11
	8	2	13	2	33	1	33
Field 2	9	2	32	2	31	1	22
Rapeseed	10	1	12	1	22	2	11
	11	1	13	1	33	2	13
	12	1	21	1	21	2	12
	13	1	23	1	31	2	23
	14	1	22	1	32	2	32
	15	1	11	1	13	2	33
	16	1	33	1	12	2	31
	17	1	31	1	21	2	21
	18	1	32	1	11	2	22

Coulter (C)1. With Coulter Coulter (C)2. Without Coulter

HEIGHT (H) 1. 4" WIDTH (W) 1. 0.5" 11. W (1) H (1)
HEIGHT (H) 2. 6.75" WIDTH (W) 2. 1"
HEIGHT (H) 3. 9" WIDTH (W) 3. 1.5"

used to determine if each parameter and their various interactions had an effect on pull.

The general form of analysis shown on Table 4.1, has two error terms because of the split plot design. A preliminary analysis was made on each dependent variable to determine whether error '1' used for testing the effect due to the presence or absence of coulter and replicates was significantly different from error '2' used for testing the other sources of variation. When the mean square of error "1' divided by the mean square for error '2' was less than the corresponding tabulated F-ratio at the 10 percent probability level, the difference between the two errors was not significant. In that case error '1' was combined with error '2' to form a pooled error used in testing all sources of variation (34).

Once the significant main effects and their interaction had been identified, Duncan's Multiple Range Test (34) was used to establish exactly which of the treatment levels were significantly different from each other. All multiple range tests were performed at the five percent probability level.

CHAPTER V

RESULTS AND DISCUSSION

The percentage of seedlings emerged and the pull measured during the test runs are given in Appendices 1 to 20. The emergence of seedlings was maximum on the 9th day in the case of the wheat and on the 6th day in the case of the rapeseed. The seedling emergence data on these days were noted as the middle counts.

As noted earlier, the first, middle and the last counts (with reference to seedling emergence) and the pull measured for wheat and rapeseed plots were used for analyses. Analyses of variance (Tables 5.1, 5.2, 5.3, 5.4, 5.5, 5.6, 5.7, 5.8) were carried out in accordance with the model noted in subsection 4.6. As can be seen, the main effects due to rotor blade width and disk height are significant for seedling emergence and pull. The effect of the coulter is only significant for pull. The effect due to the interaction between the disk height and rotor blade width is significant only for seedling emergence and the effect due to the interaction between rotor blade width and coulter is only significant for pull.

5.1 Seedling Emergence

The percentage of emergence as noted on the last day of the count was 78 percent in the case of the rapeseed but only 55.3 per cent in the case of the wheat. This low emergence of wheat seedlings is due to the reasons noted below.

TABLE 5.1

PULL FOR RAPESEED PLOT

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degree of Freedom	Mean Squares	F Value	Probability
W	30856.0	2	15428.0	4.1015	0.0248*
H	49329.0	2	24664.0	6.5570	0.0037*
WH	2685.2	4	671.30	0.1785	0.9481
C	175100.0	1	175100.0	46.5508	0.0000***
WC	208.33	2	104.17	0.0277	0.974
HC	6458.3	2	3229.2	0.8585	0.4333
WHC	16667.0	4	4166.7	1.1077	0.3679
Error R/WHC	135420.0	36	3761.6		
TOTAL		53			

*Significant at 5% level, **Significant at 1% level, ***Significant at 0.1% level.

TABLE 5.2

PULL FOR WHEAT PLOT

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	116940.0	2	58472.0	22.5536	.0000***
H	50833.0	2	25417.0	9.8036	0.0004***
WH	6388.9	4	1597.2	0.6161	0.6539
C	60000.0	1	60000.0	23.1428	0.0000***
WC	21944.0	2	10972.0	4.2321	0.6224*
HC	5833.3	2	2916.7	1.1250	0.3358
WHC	13056.0	4	3263.9	1.2589	0.3040
Error	93333.0	36	2592.6		
TOTAL		53			

TABLE 5.3

PERCENTAGE OF SEEDLING EMERGENCE FOR WHEAT AT THE END OF 5th DAY

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	110.12	2	55.060	29.0074	0.0000***
H	415.84	2	207.92	109.5387	0.0000***
WH	69.463	4	17.366	9.1488	0.0000***
C	0.78241	1	0.78241	0.4122	0.5249
WC	1.3981	2	0.69907	0.3683	0.6945
HC	1.8426	2	0.92130	0.4854	0.6194
WHC	4.5185	4	1.1296	0.5951	0.6684
Error	68.333	36	1.8981		
TOTAL		53			

TABLE 5.4

PERCENTAGE OF SEEDLING EMERGENCE FOR WHEAT AT THE END OF 9th DAY

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	564.04	2	282.02	101.8663	0.0000***
H	2568.6	2	1284.3	463.8989	0.0000***
WH	636.32	4	159.08	57.4608	0.0000***
C	0.0046297	1	0.0046297	0.0017	0.9676
WC	1.1481	2	0.57407	0.2074	0.8137
HC	4.5093	2	2.2546	0.8144	0.4509
WHC	3.3796	4	0.84491	0.3052	0.8726
Error	99.667	36	2.7685		
TOTAL		53			

TABLE 5. 5

PERCENTAGE OF SEEDLING EMERGENCE FOR WHEAT AT THE END OF 14th DAY

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	1431.6	2	715.78	75.7888	0.0000***
H	6160.7	2	3080.3	326.1548	0.0000***
WH	2802.4	4	700.61	74.1822	0.0000***
C	1.3380	1	1.3380	0.1417	0.7088
WC	11.231	2	5.6157	0.5946	0.5571
HC	29.481	2	14.741	1.5618	0.2238
WHC	40.324	4	10.081	1.0674	0.3868
Error	340.00	36	9.4444		
TOTAL		53			

TABLE 5.6

PERCENTAGE OF SEEDLING EMERGENCE FOR RAPESEED AT THE END OF 3rd DAY

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	331.84	2	165.92	127.0887	0.0000***
H	550.34	2	325.97	249.0673	0.0000***
WH	278.30	4	69.574	53.2907	0.0000***
C	0.16667	1	0.16667	0.1277	0.7203
WC	3.0833	2	1.5417	1.1809	0.3186
HC	5.2500	2	2.6250	2.0106	0.1487
WHC	4.1667	4	1.0417	0.7979	0.5345
Error	47.000	36	1.3056		
TOTAL		53			

TABLE 5.7

PERCENTAGE OF SEEDLING EMERGENCE FOR RAPESEED AT THE END OF 6th DAY

ANALYSIS OF VARIANCE

Source of Variation	Sum of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
W	182.84	2	91.421	6.9167	0.0029**
H	3378.0	2	1689.0	127.7827	0.0000***
WH	442.05	4	110.52	8.3615	0.0001***
C	5.0417	1	5.0417	0.3814	0.5407
WC	11.083	2	5.5417	0.4193	0.6607
HC	9.3611	2	4.6816	0.3541	0.7042
WHC	35.389	4	8.8472	0.6694	0.6175
Error	475.83	36	13.218		
TOTAL		53			

TABLE 5.8

PERCENTAGE OF SEEDLING EMERGENCE FOR RAPESEED AT THE END OF 9th DAY

ANALYSIS OF VARIANCE (SPLIT PLOT DESIGN)

Source of Variation	Sums of Squares	Degrees of Freedom	Mean Squares	F Value	Probability
R	219.23	2	109.62	0.9083	0.5240
C	84.375	1	84.375	0.6992	0.4911
Error (1)	241.36	2	120.68		
SUB TOTAL (1)		5			
W	1327.4	2	663.70	14.5112	0.0000***
H	392.93	2	196.46	4.2955	0.0212*
WH	578.63	4	144.66	3.1628	0.0251*
CW	22.694	2	11.347	0.2481	0.7816
CH	70.778	2	35.389	0.7737	0.4688
CWH	258.28	4	64.569	1.4118	0.2498
Error (2)	1646.4	32	45.737		
TOTAL		53			

The slippage of the walking wheels (which was not measured) was one of the main problems encountered during the test. The rotor that receives power from the walking wheel did not turn during slippage. Hence during slippage no additional 'U' shaped furrow was formed. As a consequence the seeds fell on the botton of the furrow opened by the disks only.* The rapeseed being small in size could manage to get adequate soil coverage required for germination and emergence.

In addition to this half of the rapeseed plot has mulch coverage which reduced the problem of slippage.

5.1.1 Main Effects

Effect of 'rotor blade-width' on 'seedling emergence.' The relationship between rotor blade width and percentage of seedlings emerged is displayed in Figure 23. In general there was maximum emergence when the rotor blade width was 1.0 in. When the rotor blade width was 0.5 in., the base width of the 'U' shaped furrow opened by the rotor was small. The furrow was closed by soil, falling from its wall before the seeds fell; hence most of the seeds were on the top of the furrow, rather than falling on the furrow bottom. This gave rise to a very little soil coverage on the seeds. A rotor blade of 1.5 in. width opened a furrow of considerable large width. Some of

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*When the machine was used without coulter slippage of walking wheels occured and a ridge of uncultivated soil was sometimes left on the middle of the furrow. This is displayed in Figure 23.

Figure 22. Ridge of Uncultivated Soil During Slippage of Walking Wheel

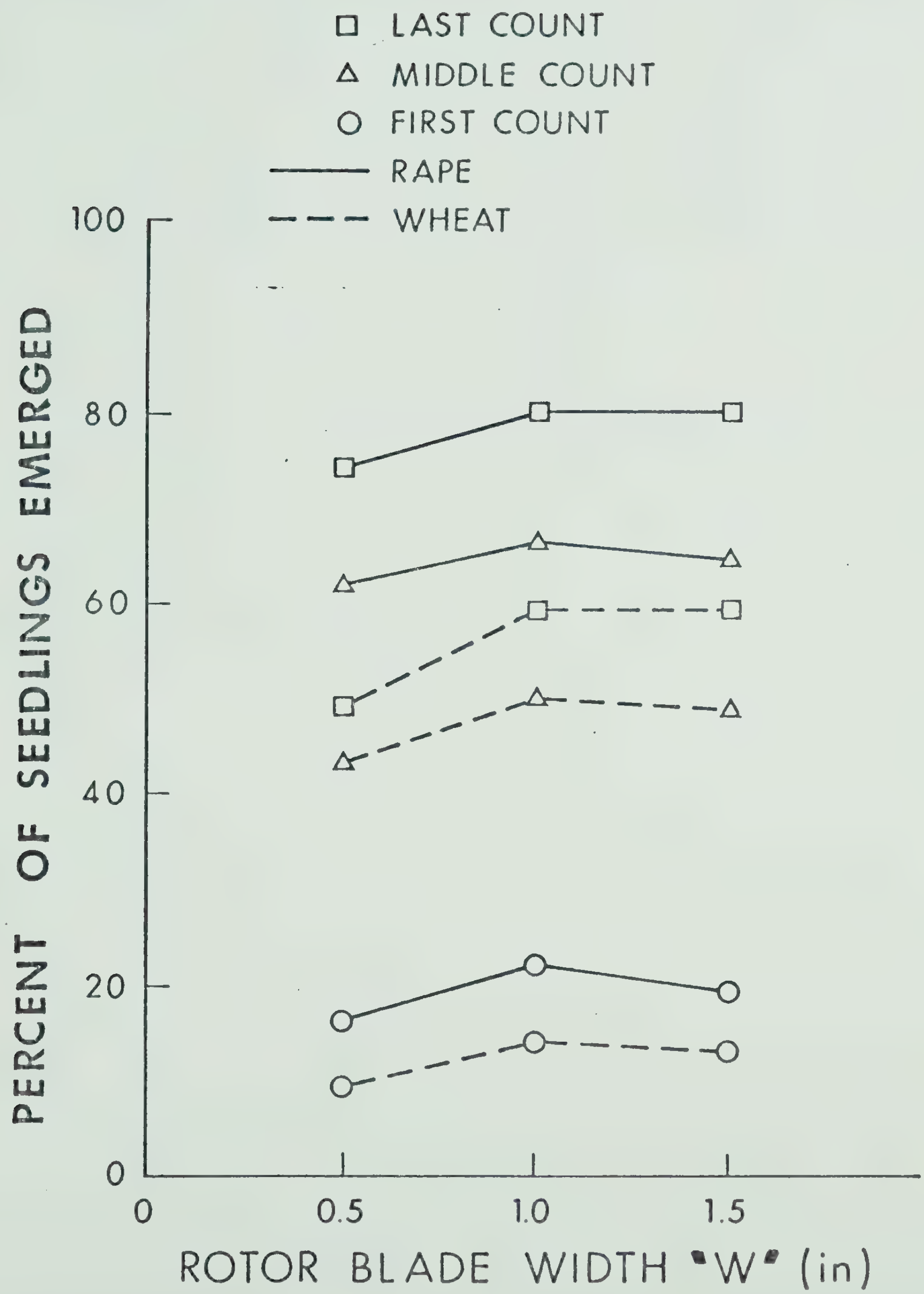


Figure 23. Effect of 'rotor blade width' on 'seedling emergence'.

the seeds in this case were displaced in either direction from the center line of the furrow giving rise to unequal heights of soil coverage on the seeds. Seeds that fell close to the furrow wall were often covered with soil to a height more than 1.5 in. to 2.0 in.; hence they could not emerge. On the contrary a rotor blade of 1.0 in. width opened a cleaner furrow, in comparison to the other two rotor blades and deposited the seeds under an ideal height of soil coverage. Therefore the percentage of seedling emerged was maximum when a rotor blade of 1.0 in. width was used in the machine.

Effect of 'height at which disks unite' on 'seedling emergence.' Effect of 'H' on the percentage of seedlings emerged in shown in Figure 24. Seed germination was poor and consequently the seedling emergence was low when the disks met at a height of 4.5 in. The reason was that the disk angle being negative, the inversion of furrow slice was poor. Hence most of the soil fell back into the furrow before the seeds reached the furrow bottom. As a result, the seeds got covered by dry soil and also the soil coverage on the seeds became thin. When the disks united at a height of 9.0 in. there was a ridge of soil left after the disks. This soil was loosened when the coulter was attached to the machine and stayed compact when the coutler was not attached to the machine. In either case, whether the ridge was broken or not, the soil was displaced by the action of the rotor alone. The rotor opened

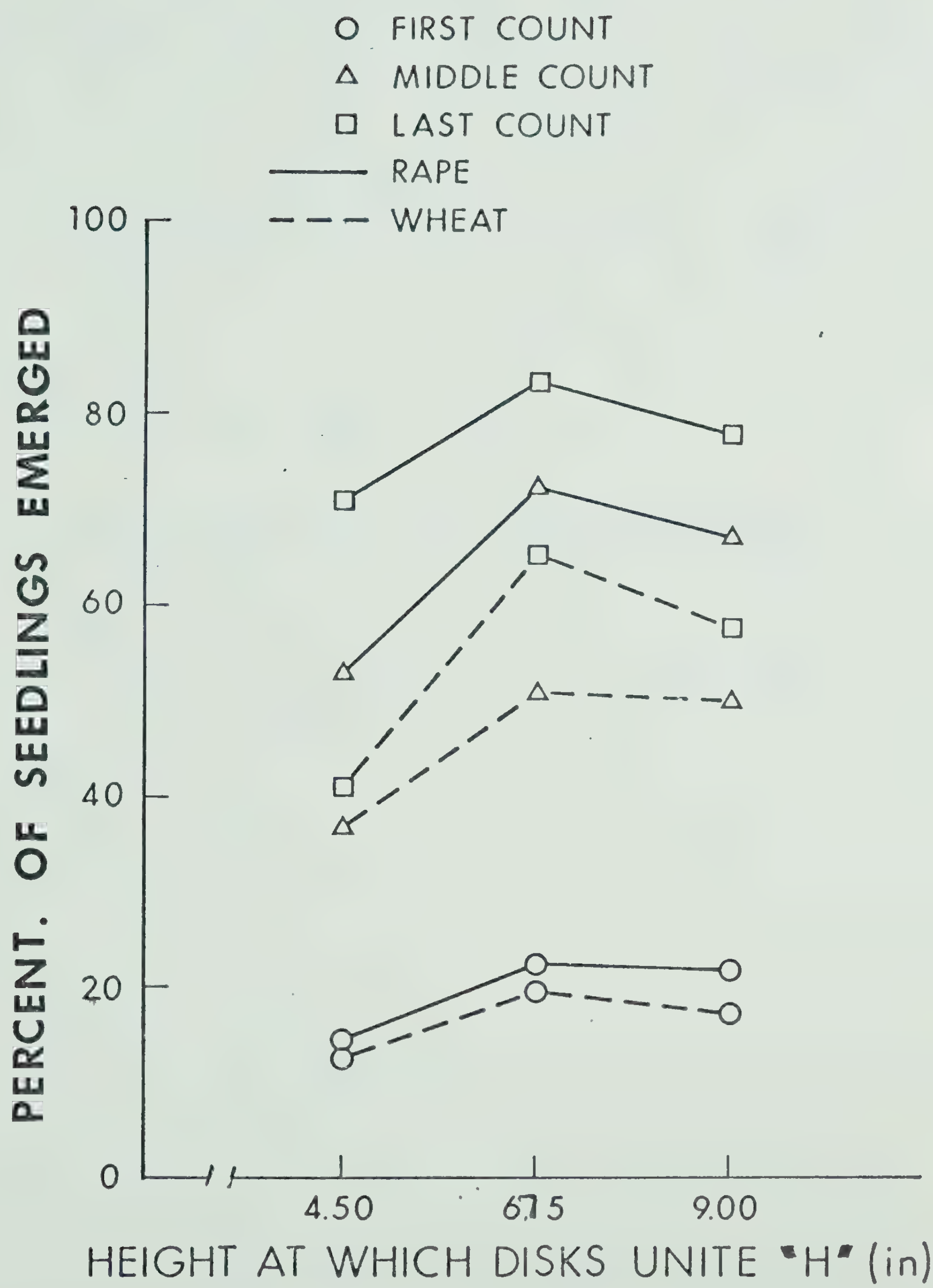
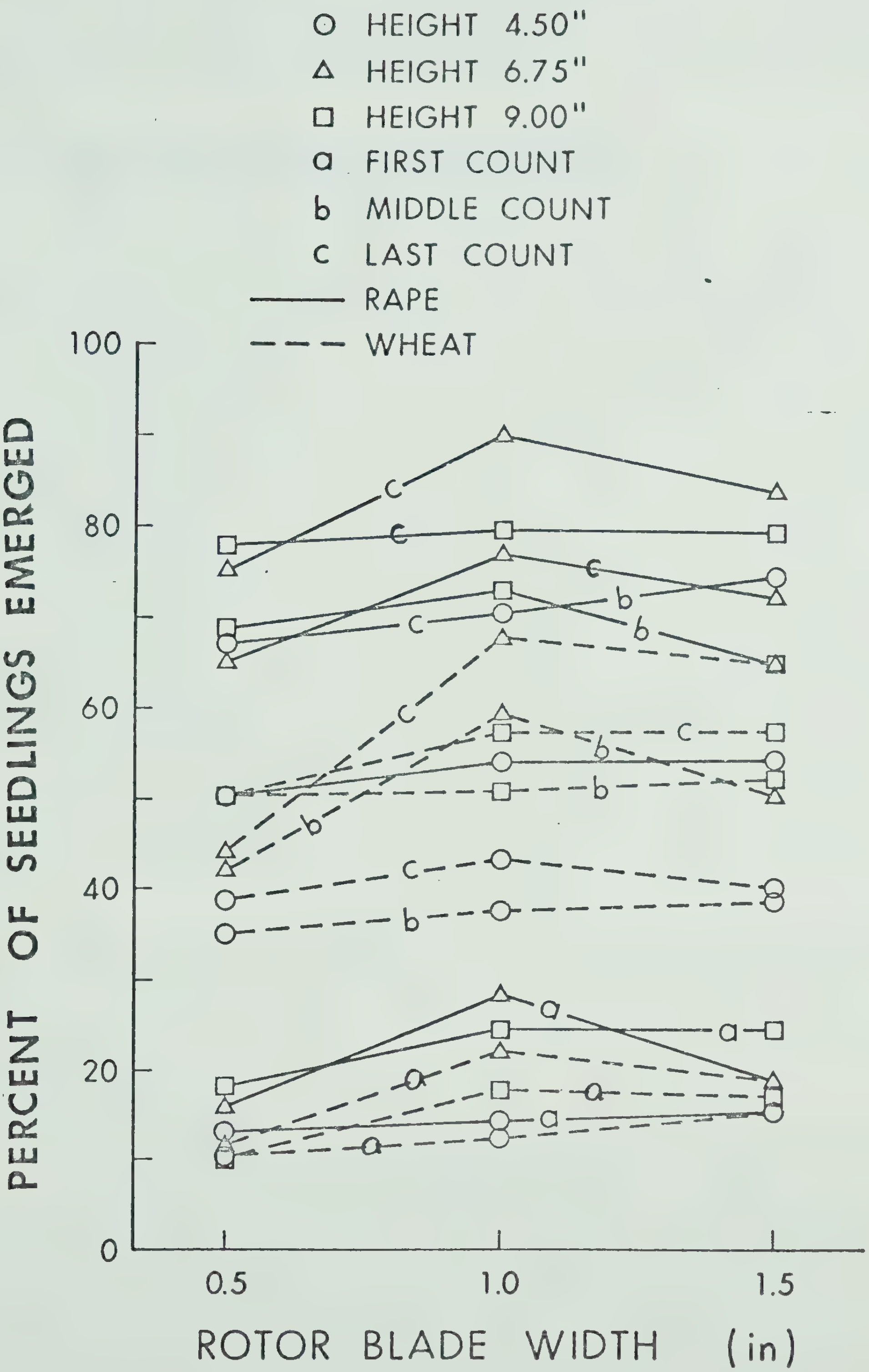


Figure 24. Effect of 'disk height' on 'seedling emergence'.

a 'U' shaped furrow, nearly 4 to 6.0 inches in height. When no coulter was used and the direction of the machine was not a straight line because of the slippage of one wheel, some part of the ridge stayed unbroken even after the rotor. On the other hand, when the coulter was used, due to unusual furrow shape, most of the soil fell back into the furrow, before the seeds reached the furrow bottom. A compromise was obtained between the poor inversion of furrow slice and unusual furrow shape, therby opening a more clean furrow and covering the seed with approximate ideal amount of soil in the case of 6.75 in. disk height.

5.1.2 'Disk height' and 'Rotor blade width'Interaction

The interaction between the rotor blade width and disk height is displayed in Figure 25. The reason for maximum percentage of seedling emergence at a rotor blade width of 1.0 inch has already been explained under 'Main Effects.' In all three counts for a disk height of 4.5 in. or 9.0 in., seedling emergence percentage between the use of a rotor blade of 1.0 in. width or 1.5 in. width remained more or less same. In fact, a rotor blade of 1.5 in. width opened a furrow which was a little more cleaner than the other and hence, there is a slight upward tendency of the curve from 1 to 1.5 in. showing a slight increase in the seedling emergence percentage in the case of latter. The same nature of the curve although to a lesser extent, is envisaged between a rotor blade width of 1.0 in. and 1.5 in., in the case of all 3 counts for a disk



('H'x'W' interaction)

Figure 25. Effect of 'HxW interaction' on 'seedling emergence'.

height of 9.0 in. The reason is, rotor blade of 1.0 in. width or 1.5 in. width made very little difference in practical observation in opening an 'U' furrow of up to 6.0 in. in height.

5.2 Energy Requirements of the Machine (In terms of the Pull Measured)

In the case of the rapeseed plot, no interaction was marked between any independent tool variables whereas in the case of the wheat plot, interaction at 5 per cent significance level was found between the width of the rotor blade and presence or absence of the coutler. This can be attributed to the experimental error arising out of two problems.

(a) Nearly half of the rapeseed plot, as has been mentioned earlier, had mulch coverage which reduced the problem of slippage. Consequently pull requirement in this case was lower in magnitude than the pull when slippage was encountered during the operation of the machine in the wheat plot.

(b) The dynamometer, due to rapid fluctuation of the pointer sometimes was read inaccurately.

5.2.1 Main Effects

Effects of 'rotor-blade-width' on 'pull.' This effect has been displayed in Figure 26. As the rotor blade width was increased, the volume of soil displaced by the rotor increased, so also the energy requirement of the rotor, thereby contributing to the total pull of the machine. There was a sharp increase in pull for rotor width from 0.5 inch to 1.0 inch in

the case of the wheat plot. This is logical since energy requirement is directly proportional to the volume of soil displaced.

<u>Effect of 'disk height' on 'pull'.</u> The effect of disk height on pull is diplayed in Figure 27. At a 'H' value of 4.5 inches, the tilt angle is negative. As the tilt angle was increased, more and more soil came in contact with the bearing area of the disk and consequently the draft requirement of the disks increased.

<u>Effect of 'coulter' on 'pull'.</u> Contrary to the assumptions made in the design of the machine, it was found out that the addition of a coulter to the machine decreased pull. This is due to the following reasons.

(a) The slippage of the walking wheel was less pronounced in the treatments when the coulter was added. The coulter penetrated to a depth of 4.0 inches to 6.0 inches and guided the machine in a straight path (slippage in most cases was associated with the fact that the direction of travel of the machine was not straight).

(b) The coulter used to break the soil ahead of the disk, initiating less energy requirement for the disks as well as for the rotor. This was due to the fact that coulter needs less energy than what the disks or rotor need to perform the same job. The result is an overall decrease in pull.

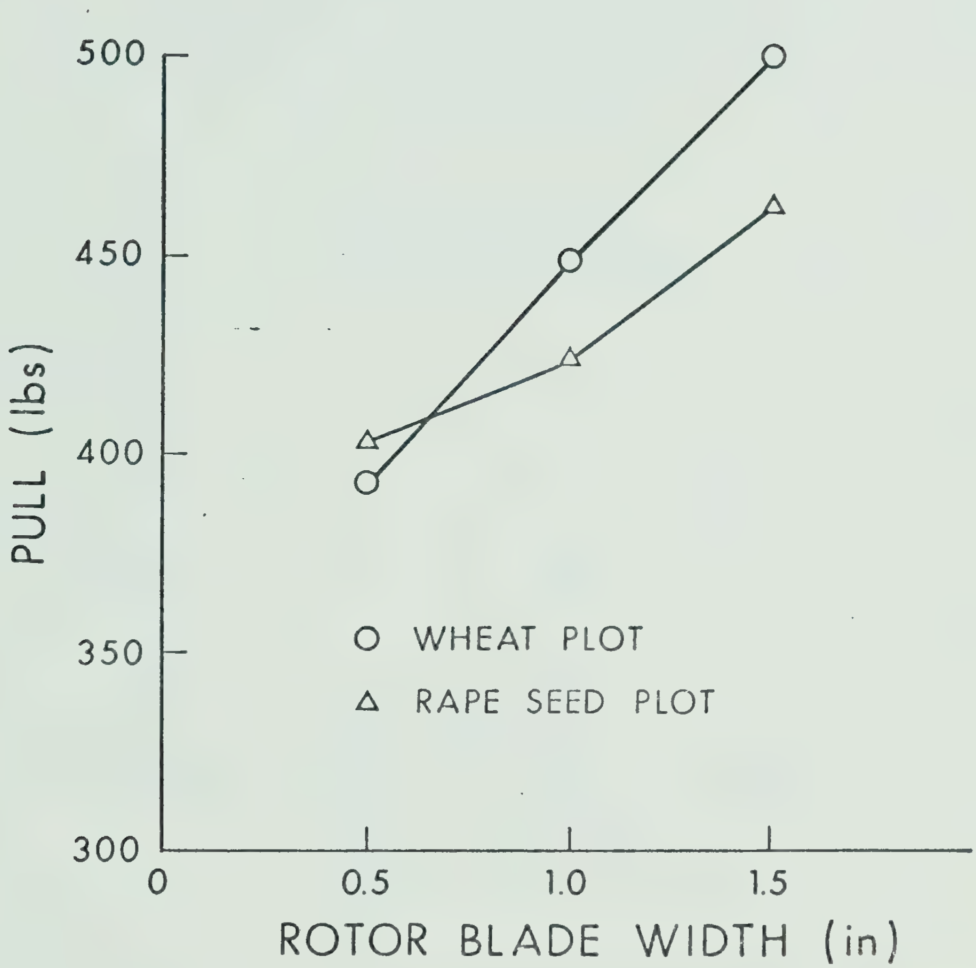


Figure 26. Effect of 'rotor blade width' on 'pull'.

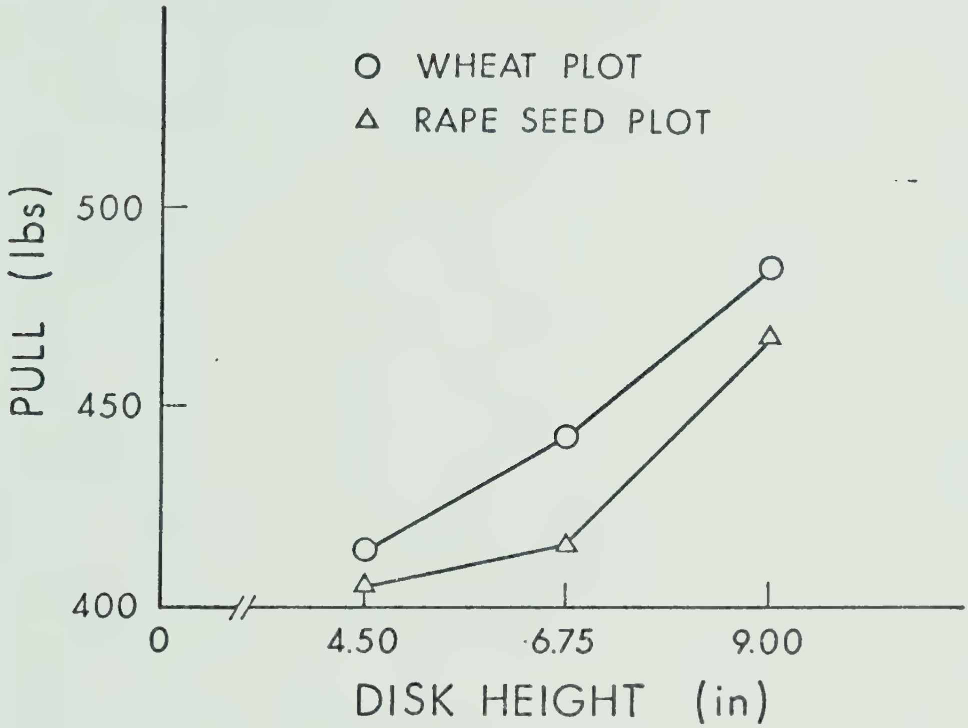


Figure 27. Effect of 'disk height' on 'pull'.

5.2.2 'Coulter' and 'rotor blade width' Interaction

The interaction of 'W' and 'C', which was only marked in the case of the wheat plot is displayed in Figure 28. The increase in pull due to increase in rotor blade width and the decrease in pull due to the presence of the coulter have been explained under 'Main Effects.'

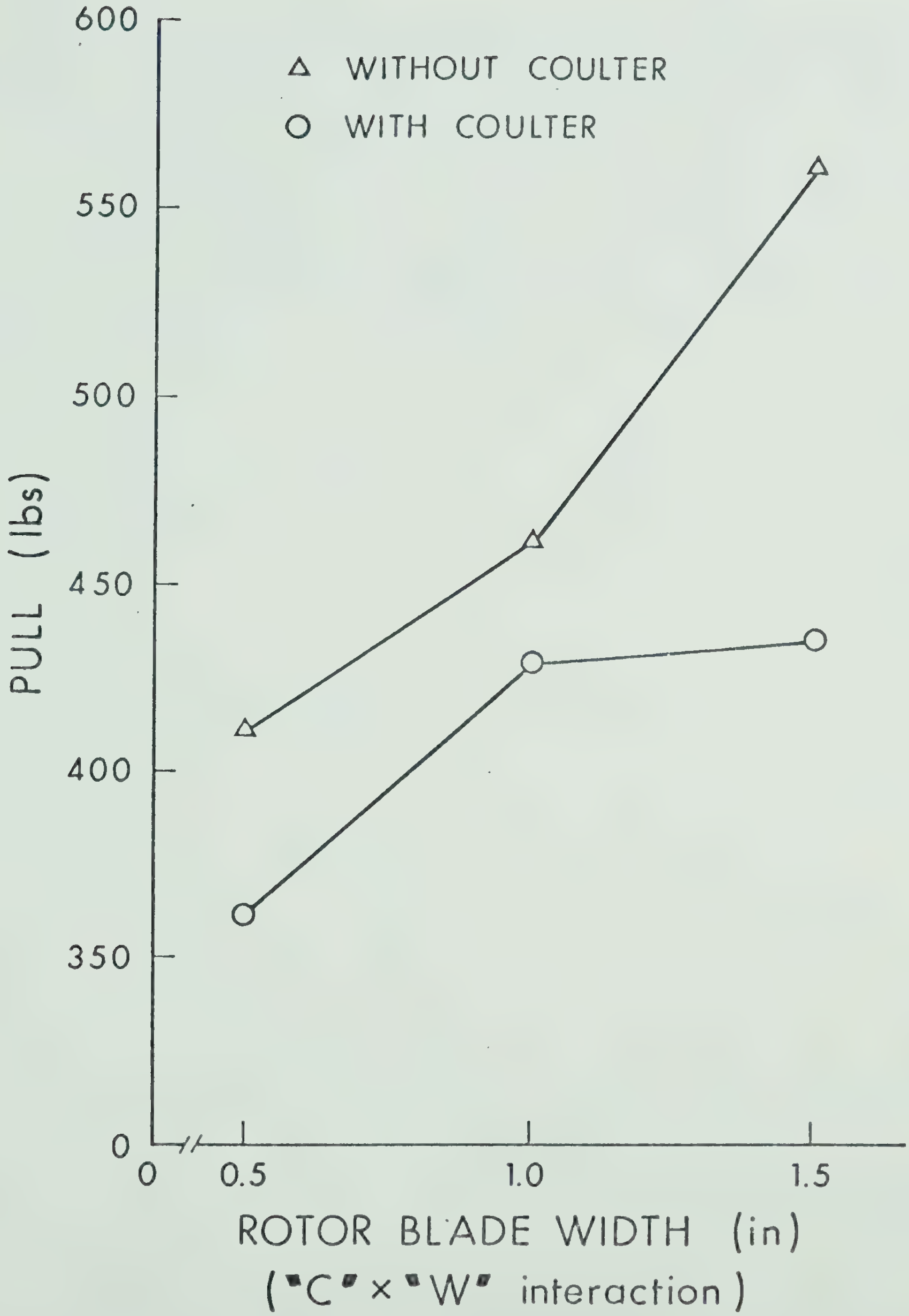


Figure 28. Effect of 'C x W interaction' on 'pull'.

CHAPTER VI

SUMMARY AND CONCLUSIONS

The performance of the experimental furrow opener unit shows that the maximum seedling emergence in all three counts was obtained for a combination of rotor blade width of 1.0 in. and a disk height of 6.75 in. Although the coulter had no effect on the seedling emergence either in the case of the wheat or rapeseed, by attaching a coulter the energy requirement of the machine was reduced. The energy requirement of the machine was, however, not minimum for a combination of rotor blade width of 1.0 in. and a disk height of 6.75 in. The pull was minimum for a rotor blade width and disk height combination of 0.5 in. and 4.5 in. respectively and was maximum for a rotor blade and disk height combination of 1.5 in. and 9 in. respectively.

The mean pull needed by the machine with the coulter attached and for a treatment combination of rotor blade width of 1.0 in. and a disk height of 6.75 in. was 450 lbs. for wheat plot and 366 lbs. for rapeseed plot. The dynamometer reading being approximate, the pull can be assumed to be 450 lbs. in which case the horsepower requirement of the machine is nearly 1.8 h.p. The mean pull needed without the coulter and for the above treatment combination was 350 lbs. for wheat plot and 333 lbs. for rapeseed plot. The maximum of

the two values is 350 lbs. and should be taken into consideration for the purpose of design. Based on this, h.p. requirement of the machines is 1.4.

The percentage of emergence (final count) of wheat for a treatment combination of 0.5 in. and 4.5 in., for rotor blade width and disk height respectively, was approximately 39 (effect of coulter being insignificant). The same for rapeseed was 68 percent. The emergence (final count) of rapeseed for a treatment combination of 1.0 in. and 6.75 in. for rotor blade width and disk height respectively was 91 percent. The same for wheat was 78 percent (the effect of coulter being insignificant for both the cases). Increasing the machine h.p. requirement by 30 per cent the gain in seedling emergence is 95 percent in the case of wheat and 33 percent in the case of rapeseed. Hence it is logical to conclude that the independent tool parameters that will give best performance as far as pull and seedling emergence percentage are concerned are

(1) presence of coulter

(2) rotor blade width of 1.0 in.

(3) disk height of 6.75 in.

In addition, an important conclusion can be drawn that the machine with the use of the above design parameters can be effectively used for dryland seeding operation in India. The horsepower requirement of this machine is 1.8 h.p. (when the depth of seed placement is 6 in.) which can be easily provided by two pairs of bullocks.

CHAPTER VII

SUGGESTIONS FOR FURTHER RESEARCH

1. Conclusions reached in this study from the test results of the machine should not govern the optimum design parameters needed for commercial production. Further test of the machine in a dryland area with sandy loam soil is essential for the selection of the optimum tool variables.

2. A triple disk combination followed by a hoe should be included in the test for actual comparison.

3. Steel or rubber lugs should be attached to the walking wheel of the machine to avoid slippage.

4. During further tests with this machine, some provisions should be made to guide the soil (cut by the rotor blade and thrown ahead) back into the furrow after the seed spout. This is required to save other parts of the machine from getting plugged up with wet soil.

5. Soil removing blades should be attached to the concave side of each disk.

6. A spring dynamometer is not an ideal tool to measure the pull because of rapid fluctuation of pointer. Instead a strain gauge unit coupled to a recorder will avoid many erroneous readings.

7. Uniformity of seed-fall through the spout also affects the germination rate. A seed metering device should be

enclosed with the machine, in which case the frames of the machine should be modified to be lighter in weight than the existing heavy frame; so that the total weight of the machine stays unchanged.

8. The machine should be tested at other depths of operation, ranging from 2 inches to 6 inches.

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APPENDIX I

MECHANICAL DESIGN AND PARTS

A.1 Design of Disk Housing

Approach

(i) Two disks of 18 inch diamter, and 1.75 inch concavity were selected. The pitch circle radius was calculated. The disks with their concave sides facing outward were allowed to meet at a point in their periphery with the minimum disk angle provided.

(ii) Soil-disk reactions were determined from experimental findings.

(iii) The bearings and the axles were designed.

(iv) The axles were bolted into a 3 inch diameter mild steel solid hub, in such a way the axes of axles lie in one plane and make and agle of (180^0 - 2 x disk angle) in between.

Theory and Calculations

Radius of Curvature

The radius of curvature, shown in Figure 29 is given by the equation:

$$d = \frac{\sqrt{4R^2-c^2}}{2}$$

Where R = pitch circle radius or radius of curvature

= (d+h),

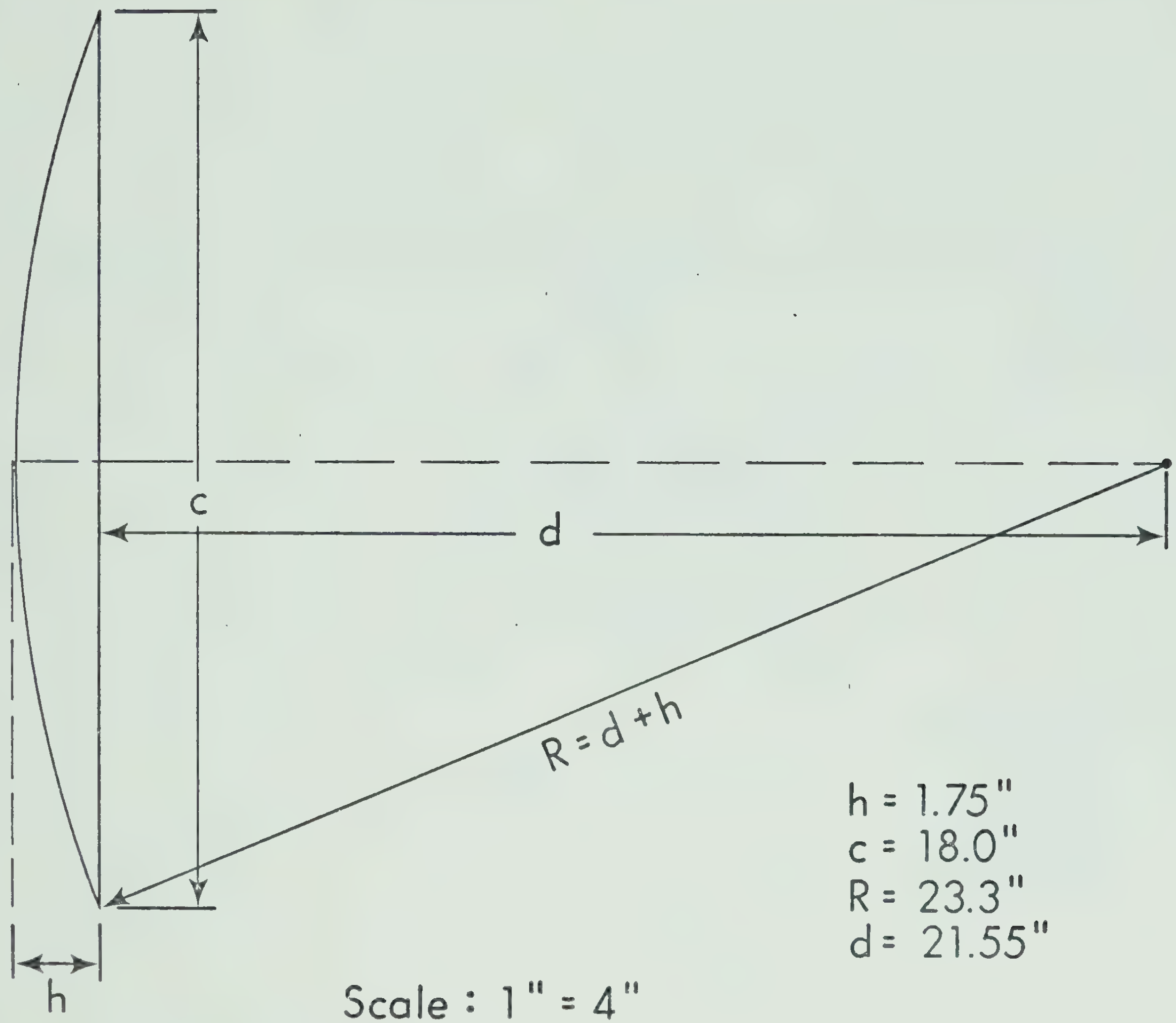
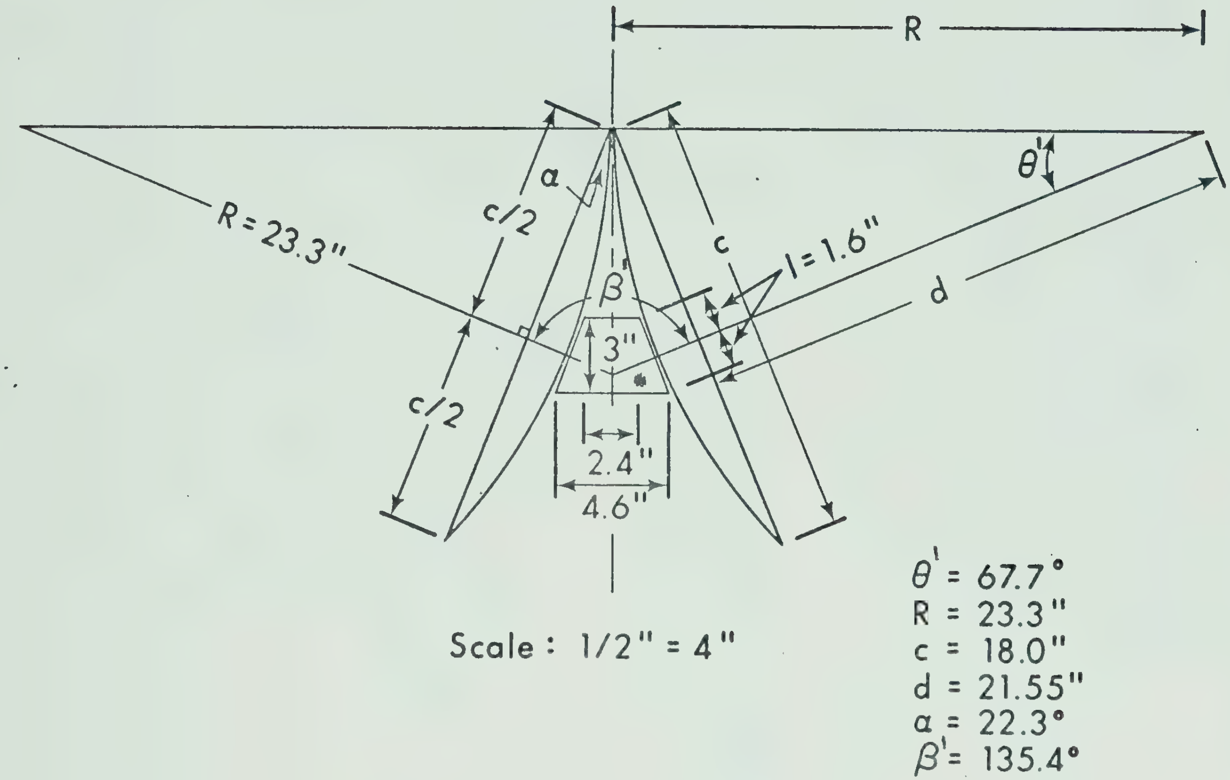


Figure 29. Disk Showing the Radius of Curvature.



*Disk Housing

Figure 30. Double Disk Unit Showing Minimum Disk Angle.

h = concavity of disk; c = diameter of disk or,

$$2d^2 = 4R - c^2$$

$$= 4 \quad (d+h)^2 - c^2$$

or, $c^2 = 4h^2 + 8dh$

Putting the values of c and h in eq. 1,

$$(18)^2 = 4\ (1.75)^2 + 8dh\ (1.75)$$

or, $324 = 12.25 + 14dh$

or, $d = 21.55$

Therfore, $R = 21.55 + 1.75$

$= 23.30$ inches

Radius of curvature of disk is 23.30 inches

Maximum <u>angle between two disk axles, when the disks join at the periphery</u>.

Referring to Figure 30,

$$\text{Sin}\ \theta^1 = 21.55/23.30 = 0.9249$$

Therfore $\theta^1 = \text{Sin}^{-1}\ (0.9249) = 67.7^0$

Minimum disk angle $= 90^0 - 67.7 = 22.3^0$

Maximum angle between two disk angles,

$$= 67.7 \times 2 = 135.4^0$$

<u>Bearing for the disks</u>. Gordon (17) reported that no substantial difference could be observed as far as bearing friction is concerned: (1) when the disk is mounted on a spindle which rotates in an anti-friction bearing and (2) when the disk spindle rotates in a well constructed and ad-

justed plain bearing. So, for experimental purpose a housing (displayed in Figure 30) containing the ball bearing and an axle that goes through the bore of the bearing, was bolted to the concave side of the disk. The disk rotates freely on the axle.

When the disk moves in soil, there are 3 forces to take into consideration. They are: draft 'L', which is in the direction of movement of implement; the side force 'S' that acts in a direction perpendicular to 'L'; and the vertical soil reaction 'V' which is in a direction perpendicular to both 'L' and 'S'.

Clyde (10) presented two methods to express the resultant effect of the 3 forces which govern the bearing design. One of these methods assumes that 'L' and 'S' are intersecting and 'V' is nonconcurrent. This method is explained in Figure 31 b. Clyde also reported that it would be equally permissible to consider that 'L' and 'V' intersect and 'S' is nonconcurrent (10).

Clyde's second assmption displayed in Figure 31a and 32 has been taken into consideration to govern the present design of the bearing.

Gordon (17) found out the following values of 'L', and 'S' and 'V' in an experiment carried out at the United States Department of Agriculture (U.S.D.A.) Tillage Laboratory.

When (1) Nominal disk diameter = 20 inches

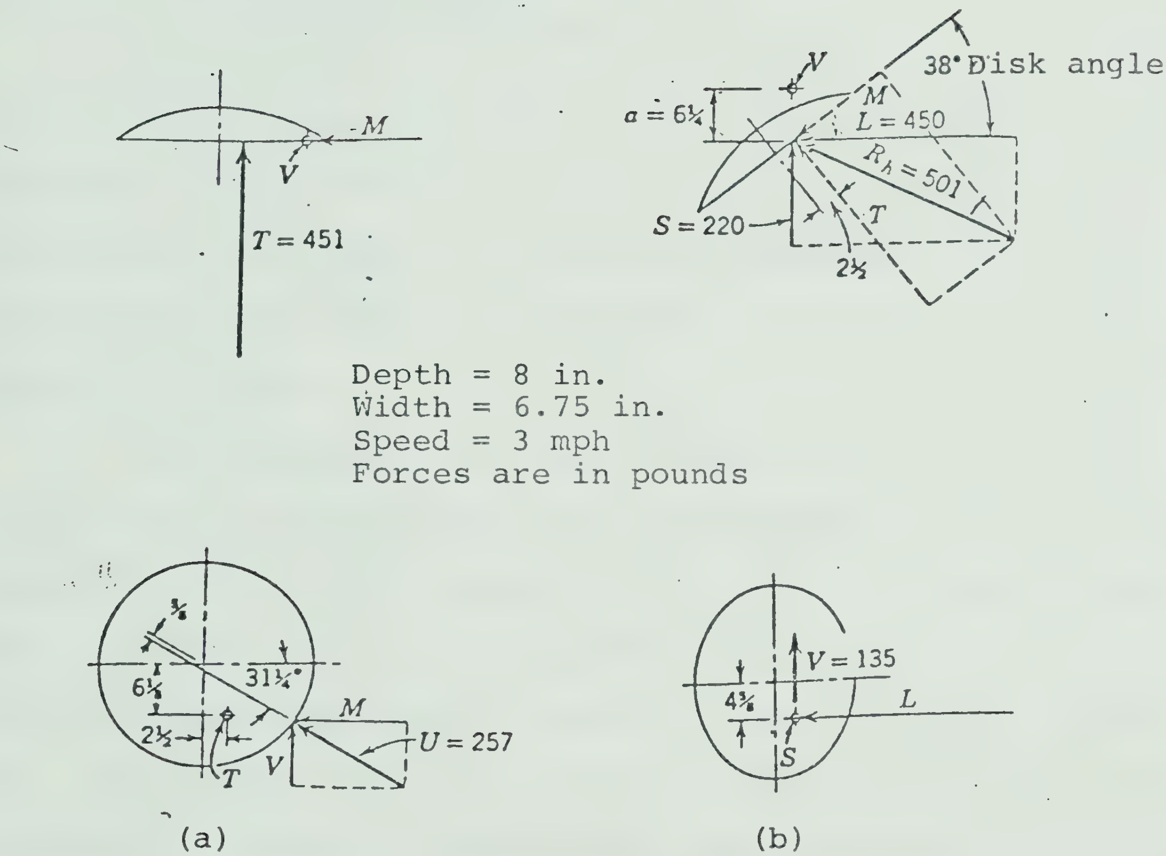


Figure 31. Example of resultant soil forces acting upon a 24 in. vertical disk blade under field conditions in a silt loam soil, the total effect being represented by 2 non-intersecting forces:

(a) a thrust force 'T' plus a radial force U, and

(b) a horizontal force Rh, plus a vertical force V (Clyde) (10).

(2) Disk angle = 45^0,

(3) Depth of cut = 4.6 inches,

(4) Width of cut = 7 inches,

(5) Angle of inclination = 0^0,

Draft L (in lbs.) = 203

Vertical soil reaction 'V' upward (in lbs.) = 60

Side thrust S (in lbs.) = 138

The soil used in the experiment was Davidson Loam that consisted of 72.8 per cent of sand, 3.4 percent of silt and 23.8 percent of clay as determined by mechanical analysis. Apparent specific gravity of soil was 1.71 and moisture content 7.45 percent.

Agricultural Engineers Yearbook (9), shows that for a disk plow in a light soil the draft requirement is 3 to 6 lbs. per square inch and for medium soil it is 5 to 9 lbs per square inch. Davidson loam can be considered as a medium soil in which the draft requirement can be assumed to be 7.0 lbs per square inch. Hence, the draft for a furrow section of 7.0 in. x 4.6 in. is 225.4 lbs, which is close to the value obtained by Clyde (10) in an experiment. (The furrow section will be approximately rectangular instead of being triangular as in a double disk furrow). Clyde (10) in the same experiment found out that, the draft 'L', vertical soil reaction 'V' and the side thrust 'S', all were lower for a disk of 26 inches in diameter than a disk of 20 inches in diameter, at the same depth of operation.

Thivavarnvongs (32) in an experiment with disks found no appreciable difference in values of 'L', 'V' and 'S' when the disks of 18 inch, 20 inch and 22 inch diameters were used.

So with a disk angle of 45^0 and tilt angle of 0^0, it can be logically assumed that the vlaues of 'L', 'S' and 'V' will not appreciably differ between a disk of 18 inch diameter and 20 inch diameter.

Plow disks are generally operated with a disk angle of 45^0 (2). Gordon (17) found that decreasing the disk angle from 45^0, the draft as well as vertical soil reaction 'V' increases but the side reaction 'S' decreases. He reported that draft is minimum at about a 45^0disk angle. Thivavarnvongs (32) also reported that 'L' attains a minimum in the vicinity of 30^0. The USDA tests, as reported by Gordon (17) indicated that soil type and soil condition have the most pronounced effect on soil reactions. There is no report available for a disk of 18 inch diameter, with a disk angle of 22.3^0 and tilt angle of 0^0, operating at a depth of 4.5 inches, in a sandy loam soil with the width of cut being 7.5 inches. Hence, with the above soil and tool conditions, for the purpose of mechanical design, 'S' can be assumed to be 138 lbs. Since 'V' increases rapidly with the lowering of disk angle as reported by Gordon (17) the value of 'V' in the above case can be assumed to be (60x2=) 120 lbs. Since there is no sharp increase in draft by decreasing the disk angle, 'L' can be assumed to be (203 x 1.5 =) 305 lbs.

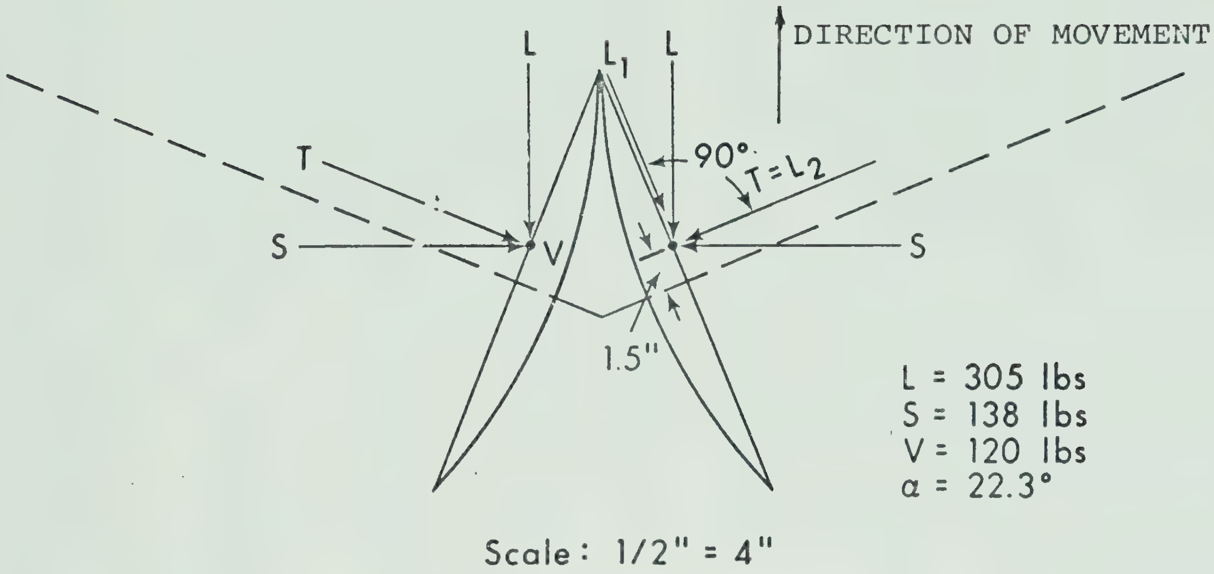


Figure 32. Direction and Magnitude of Resultant Forces on a Double Disk Unit.

(L is resolved into two components 'L_1' and 'L_2')

The magnitude and direction of 'L', 'S' and 'V' are shown in Figure 32.

L = 305 lbs.	Disk angle = 22.3° = α
S = 138 lbs.	Tilt angle = 0° = θ
V = 120 lbs.	Soil type is sandy loam
	Depth of cut = 4.0 inches

Calculation of the resultant forces is based on previous assumptions and is shown in Figure 33.

For a single disk, 'L' can be represented by two components. Component 'L_1' is parallel to the plane of the disk and component 'L_2' is perpendicular to the plane of the disk.

$$L_1 = L \cos\alpha = 305 \cos 22.3° = 281.82 \text{ lbs.}$$

$$L_2 = L \sin\alpha = 305 \sin 22.3° = 115.90 \text{ lbs.}$$

Assuming that 'L' and 'V' act at one point, the resultant of 'L_1' and 'V' can be represented by 'Ra' where 'Ra' is the radial force.

$$Ra = \sqrt{L_1^2 + V^2} = \sqrt{(281.82)^2 + (120)^2} = 306.30 \text{ lbs.} \simeq 305 \text{ lbs.}$$

The two side forces in the double disk unit cancel each other. Hence, the thrust 'T' on each disk is given by:

$$T = L_2 + 0 = 115.90 \simeq 116 \text{ lbs.}$$

The radial force which includes the vertical support force on the disk blade, must pass slightly to the rear of the disk

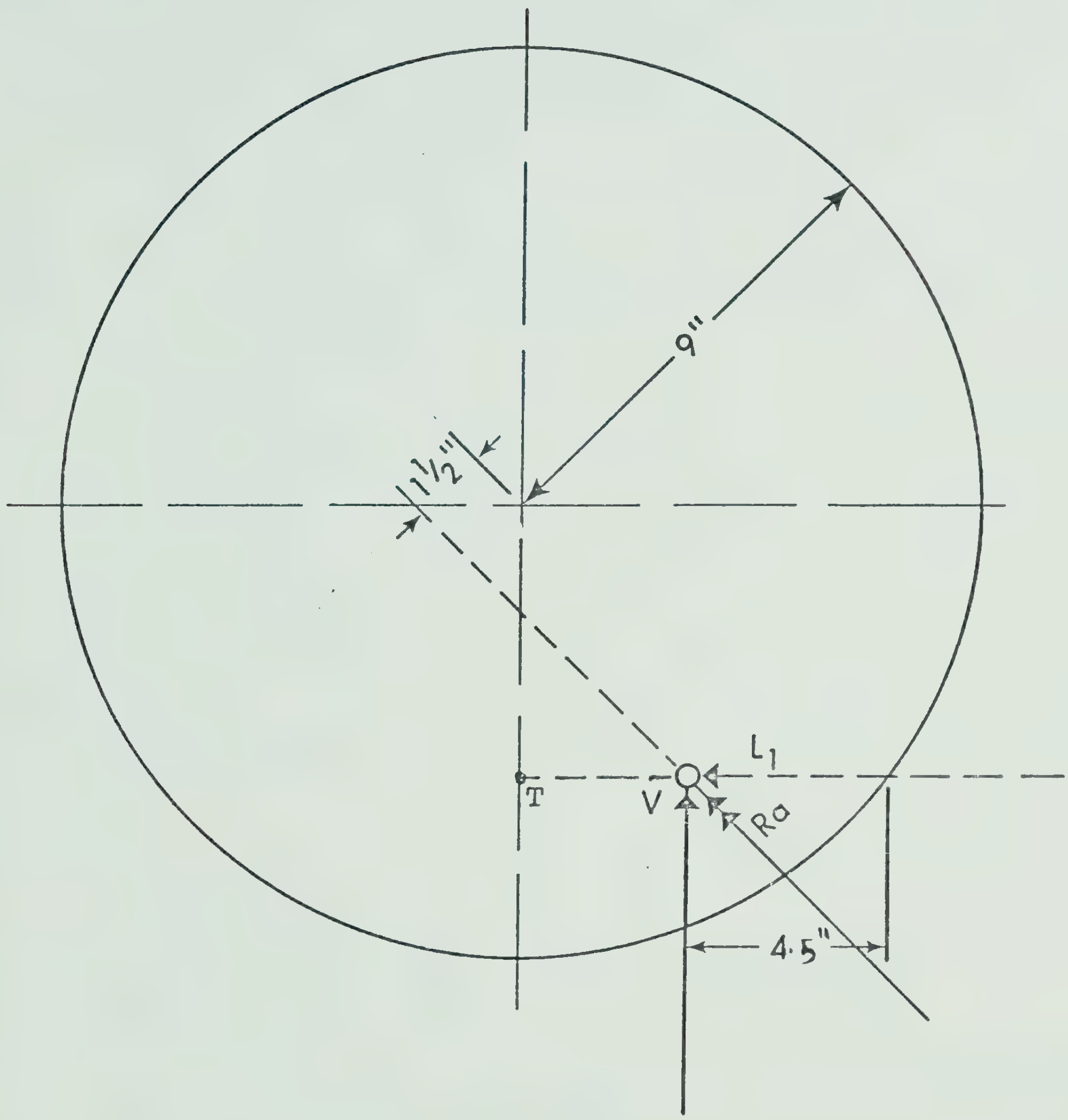


Figure 33. Resultant of 'L_1' and 'V'

center line to provide the torque mecessary to overcome bearing friction and to cause rotation of the disk. According to the report of Clyde (11) minimum distance between the line of action of the raidal force and the center line of the disk is assumed to be 1.5 inches and the position of T is assumed as shown in the Figure 33. 'L' is assumed to act at a height of 4 inches from the bottom of the disk since the depth of cut is 4.5 inches.

Applying two equal and opposite forces of magnitude 'T' at the center of the disk in a line parallel to the line of action of 'T', there is a thrust load of 116 lbs on the bearing and a couple of magnitude 48.33 ft. lbs. (116 x 5/12 = 48.33) in addition to the radial load of magnitude 305 lbs. On the disk tools the couple has a clockwise sense with right hand disks (10).

Summarizing,

Radial load on the bearing = 305 lbs.

Thrust load on the bearing = 116 lbs.

The magnitude of the couple = 48.33 ft. lb. and the sense of rotation is clockwise in a right hand disk.

To account for the couple a tapered-roller bearing is always recommended for the disk, if long life is desired (2). However, for this experimental opener unit, a ball-bearing of suitable size was designed since the lifespan is short.

The equivalent radial load is found out by using the Anti-Friction Bearing Manufacturer's Association (AFBMA) Eqn. (29).

$$Re = V.Ra \qquad \text{(Eqn. 2)}$$

$$Re = X_1.V.Ra + Y_1.T \qquad \text{(Eqn. 3)}$$

$$Re = X_2.V.Ra + Y_2.T \qquad \text{(Eqn. 4)}$$

Where, V = Rotation factor

Ra = Applied radial load

Re = Equivalent radial load

T = Applied thrust load

X = Radial factor

Y = Thrust factor

The equivalent radial load is the maximum of the three values given by Eqn. 2, 3 and 4. According to Shigley (29) for radial-contact ball bearings

When $X_1 = 1$, and when $X_1 = 0.5$,

$Y_1 = 0$ $\qquad$ $Y_1 = 1.4$

For a rotating outer ring, V = 1.2

Hence,

$$Re^{(2)} = 1.2 \times 305 = 366 \text{ lbs.}$$

$$Re^{(3)} = 1.0 \times 1.2 \times 305 + 0 \times 138 = 366 \text{ lbs.}$$

$$Re^{(4)} = 0.5 \times 1.2 \times 305 + 1.4 \times 138$$

$$= 183 + 193 = 376 \text{ lbs.}$$

Hence, the equivalent radial load that will be taken for the purpose of design is equal to 376 lbs.

To account for any shock and impact conditions to which the bearing may be subjected, an impact factor is introduced while calculating maximum radial load (31).

$$Re \quad (Max) = C_1 \; Re$$

For heavy shock $C_1 = 3.0$

Hence

$$Re \quad (max) = 3 \times 376$$

$$= 1128 \text{ lbs.}$$

Linear velocity of implement = 1.5 mile/hr.

= 132.0 ft./min.

Therefore, Linear velocity of disk = 132.0 ft./min.

R.P.M. of the disk $\simeq$ R.P.M. of the bearing

$$= \frac{132.00}{2 \times 3.14 \times 8/12} = 31.7 \simeq 32$$

Let the basic load rating be C

$$C^3 = L \times 60 \; n \; Re/10^6 \qquad \text{(Eqn. 3 \& 4) (31)}$$

where L = Expected rating life

n = rpm

Re (max) = Equivalent radial load (max)

Let L = 200,000 hrs. (average life)

$$C^3 = \frac{20,000 \times 60 \times 28 \times 1098}{10^6}$$

or $C^3 \doteq 38000$

Therefore, C = 33.75 lbs.

Referring to the Browning Catalogue, (5), a 'B100' adapter bearing with the following specifications was chosen (The bearing axle has an outside diameter of 1 inch).

Bore size = 1 inch

C = 1.36 inches, D = 2.0472 inches, M = .761 inch ,

S = 1.061 inches, W = 0.609 inch , where C, D, M, S and W are illustrated in Figure 34.

Check

Referring to the Browning Catalogue, Load rating for the above bearing for 2500 hrs. = 1595 lbs.

Load rating for 20,000 hrs. = 1595 x 0.59 = 941.0 lbs.

The above ball bearing is specified to be used with rpm of 100 and load rating of 941.0 lbs. Since the required load rating is 33.75 lbs. for 32 rpm, the bearing chosen will be satisfactory.

A.2 Rotor Blade Design

Direction of Rotation

The seed spout follows the rotor blade. For the convenience of seed placement, the direction of rotation of the rotor blade was chosen reverse to the direction of movement of machine. For the lack of a confirmed report relating draft requirement and direction of rotation of a rotary tiller, the draft requirement was not taken into consideration while selecting the direction of rotation.

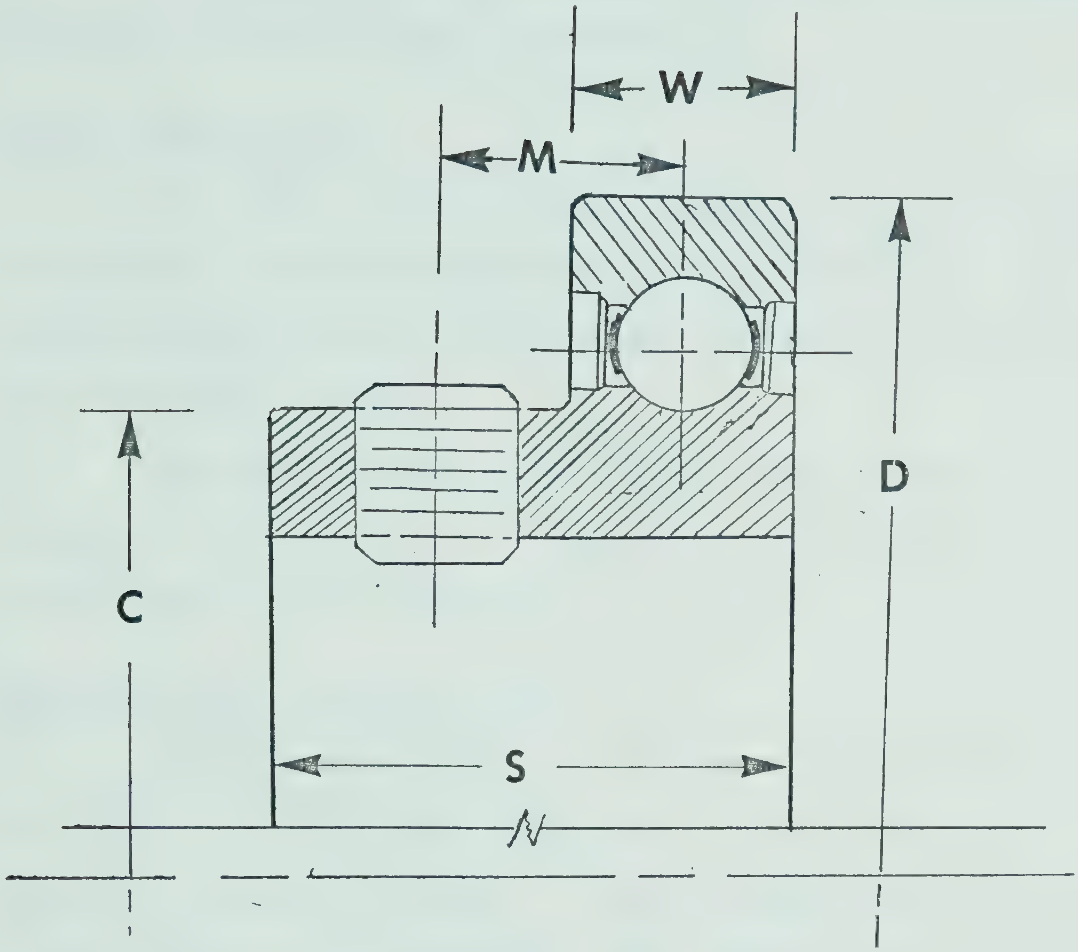


Figure 34. B-100 Adapter Bearing (half section).

C = 1.36 inches, D = 2.0472 inches, M = .761 inch

S = 1.061 inches, W = 0.609 inch.

Rotor-Blade Shape

L-shaped blade was chosen, because of its effectiveness in dryland tillage operation (1).

Rotor-Blade Length

The long arm of L shaped blade was made 8 inches long, to provide 2 inch clearance between the rotor axle and the ground surface so that the sprocket in the rotor axle does not touch the ground.

The shorter arm of the rotor blade was made 2 inches long so that the ratio of the length of the shorter arm and the length of the longer arm = 1:4 (18).

Minimum Width of Rotor Blade

The minimum width of the rotor blade is determined by taking the fact into consideration that the furrow opened should be clean and stable. The minimum width of rotor blade, taking the above fact into account was assumed to be 0.5 inch.

Maximum Width of Rotor Blade

The maximum width of the rotorblade is limited by the power available to the rotor.

Power available to the rotor

= [(Total power available to the machine)

- (Power utilized by the disks)] x 0.75

(Assuming that 25 percent of the power available to the

machine is utilized by pulling the (1) coulter (2) machine itself, when no part is lowered into the ground and (3) soil covering blades.

For a clean furrow, the width of uncultivated soil left on the top of soil surface should be zero. In this case the disks should be joined at a height of 4.5 inches since the maximum depth of penetration of the disks is 4.5 inches. Joining the disks at a height of 4.5 inches at the front, the tilt angle becomes negative and there is poor inversion of the furrow slice (2). When the disks are joined at a height of 9.0 inches and tilt angle is approximately 0^0, the inversion of the furrow slice becomes better than the former, but a ridge of uncultivated soil is left on the soil surface. Hence to compensate between the two extreme phenomena, the disks should be joined (at the front) at a height between the two heights of 4.5 inches and 9.0 inches. However the effectiveness of a clean furrow can only be studied when all these three heights are taken into cosideration.

For the purpose of determination of the soil cutting force available to the rotor, the disks should be joined at the front at a height of 9.0 inches from the bottom of the disks, since the draft in this case becomes maximum.

The furrow width is maximum when the disks join at the front at the center line.

Hence the maximum furrow width is 10 inches as shown in Figure 35.

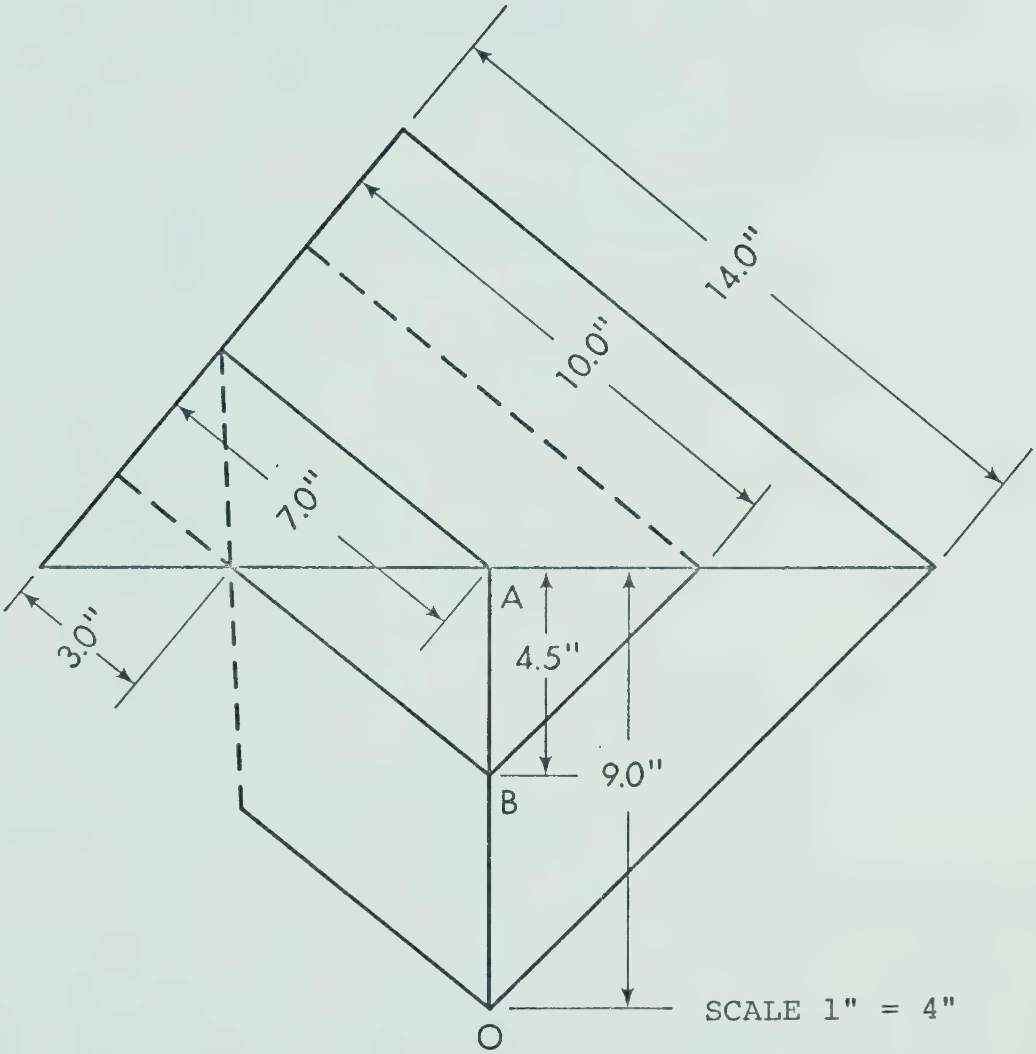


Figure 35. An Example of Determining Furrow Width When Disks are Joined at the Front.
When the disks meet at a height of 9.0 in., the width of space at the back center line is 14 in. When the disks operate at a depth of 4.5 in., the furrow width is 10 in., and the width of uncultivated soil left on the top of the furrow is 3 in.

The configuration of the furrow opened by the disks is shown in Figure 36.

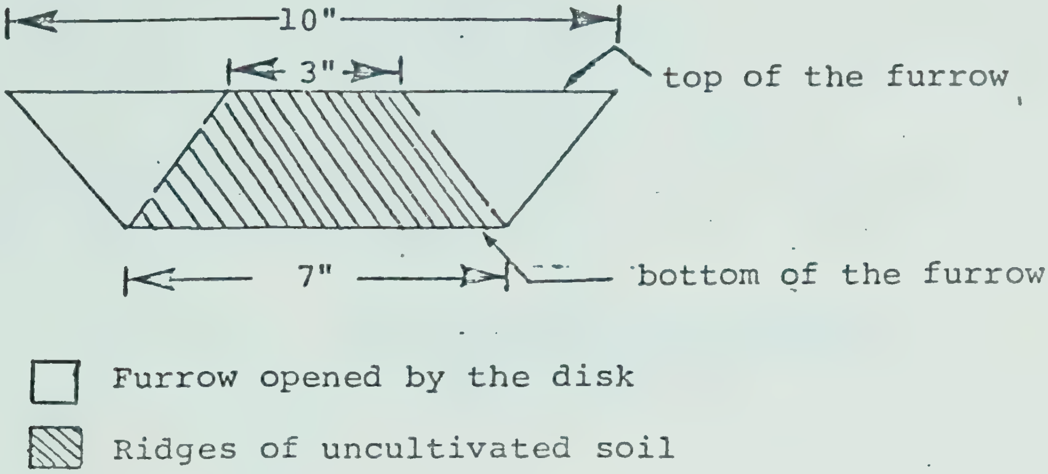


Figure 36. The configuration of the furrow opened by the disks.

Assuming that the ridge of uncultivated soil does not exist,

Cross section area of furrow

$$= 4.5 \frac{(10 + 7)}{2} = 38.25 \text{ sq.in.}$$

H.P. of a pair of medium size bullocks = 0.9 (13)

H.P. of 2 pairs of medium size bullocks = 0.9 x 2 = 1.8

Approximate bullock speed = 1.5 mile/hr. (13)

= 132 ft./min.

Velocity of disks 'V' = 132 ft./min.

Specific resistance of sandy loam = 7 lbs.sq. in. (16)

(approximately)

Hence, soil cutting force 'Fd' utilized by the disks =

7 x 38.25 = 267.75 lbs.

$$\text{Power utilized by the disks} = \frac{Fd \times V}{33{,}000}$$

$$= \frac{267.75 \times 132}{33{,}000}$$

$$= 1.07 \text{ h.p.} \approx 1 \text{ h.p.}$$

Energy requirement for the rotor is minimum at a value of

$$\lambda = 2.4 \quad (19)$$

$$\lambda = \frac{\text{Rotor peripheral speed}}{\text{Forward machine velocity}}$$

$$\text{or } 2.4 = \frac{\text{R.P.S.}}{132}$$

Assuming that pull is horizontal, (actually draft is less than the pull, since the hitch point of the machine is at a lower level than the neck height of bullock).

Power available to the rotor

$$= 0.75\,(1.8 - 1\ ) =$$

$$= (0.8 \text{ h.p.})\ 0.75 = 0.6$$

$$\text{or, } 0.6 = \frac{F \times \text{R.P.S.}}{33{,}000}$$

where F is the cutting force of the rotor

$$\text{Therefore } F = \frac{0.6 \times 33{,}000}{316.8}$$

$$= 62.5 \text{ lbs.}$$

But, F = specific soil resistance x (cross section of the furrow opened by the rotor (4).

Depth of furrow opened by rotor = 6 inches.

The specific soil resistance = 7 lbs/sq.in. (approximately).

Maximum width of furrow opened by rotor.

$$= \frac{62.5}{6 \times 7} = 1.499 \simeq 1.5 \text{ inches.}$$

Hence rotor blade widths of 0.5 inch, 1.0 inch, 1.5 inch were chosen, 0.5 inch being of minimum width which can be assumed to open a clean furrow.

Peripheral Speed of Rotor Blade

$$\lambda = 2.4 = \frac{\text{Rotor peripheral speed}}{\text{Forward velcoity of machine}}$$

$$\text{or, } 2.4 = \frac{\text{R.P.S.}}{132}$$

Therefore R.P.S. = 132 x 2.4 = 316.8 ft/min.

Sharpening Angle of Blade

Radius of Rotor = 8.24 inches = R

h = cutting depth = 6 inches

Therefore, h/R = 6/0.24 = 0.728

For h/R = 0.728, intersection angle

$$\Delta\delta = 19° \text{ (for reverse rotation)}$$

For $\lambda = 2.4$, $\gamma_0 = 35°$

$$\gamma = \gamma_0 - \Delta\delta$$

$$\text{or, } \gamma = 35° - 19°$$

$$\text{Therefore } \gamma = 16°$$

Now $\gamma = \beta + (\delta - \Delta\delta)$

or $\gamma = \beta + \delta^1$

The angle 'δ^1', must be greater than 0° to prevent the backface of the blade from compressing the uncut soil (19).

Assuming that $\delta^1 = 5^0$

$\beta = 16^0 - 5^0 = 11^0$

Thickness of Rotor Blade

Assumptions

(1) Material used: Steel AISI (American Iron and Steel Institute) 1050, Drawn at 600^0F

Fatigue strength of the material = 100 kpsi (Eqn. 6.21) (29)

(The blade fails by fatigue and not by yielding)

Assumed factor of safety = 3
Allowable stress = 100/3 = 33.3 kpsi

(2) Blade widths - 0.5 inch, 1 inch , 1.5 inches.

(3) Unit draft - 7 lbs/sq. inch.

Calculation of Thickness of Rotor Blade

(of width 0.5 inch)

Max. cross sectional area of furrow opened by rotor

= 6 x 0.5 = 3 sq. in.

Total draft = 3 x 7 = 21 ;bs.

Maximum bending moment about the neck of the rotor blade = 4 x 21 = 84 lb. inch. (Reference: Figure 37). Max. shear force = 21 lbs. (assuming that this acts at the center of furrow section) i.e. 3 inches from the bottom of the rotor blade which behaves as a cantilever beam).

$Z = bh^2/6$

Where b = width of rotor blade

h = thickness of rotor blade

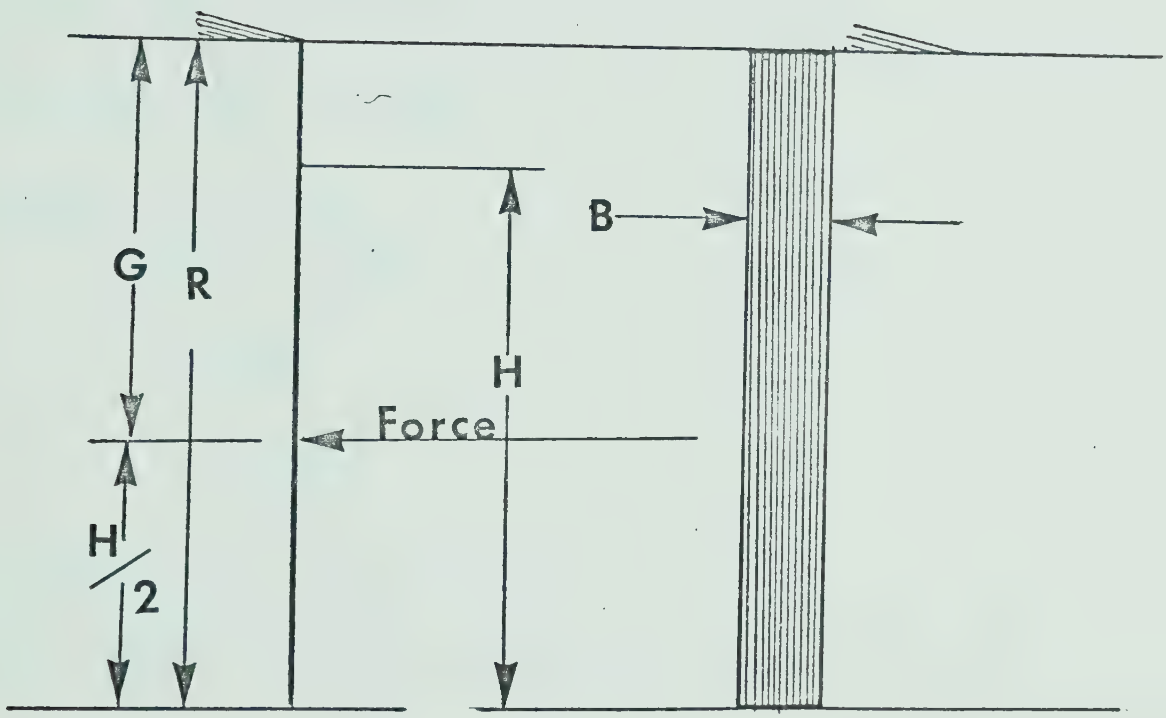


Figure 37. Force on the rotor blade.

R = Length of rotor blade = 7.0"

G = Arm of bending moment = 4.0"

B = Width of rotor blade

H = Depth of Furrow

$$Sx\ (max) = Mmax/Z$$

$$= \frac{6x\ M\ max}{bh^2}$$

Where, Sx = flexural stress

M = bending moment

Z = section modulus

$$Sxy = \frac{VQ}{Ib}$$ where, Sxy = shear stress

$$Sxy\ (max) = \frac{3V}{2A} = \frac{3 \times 21}{2 \times bh},$$

$$(Ss)\ max = \tfrac{1}{2}\ Sx\ max$$

$$= \tfrac{1}{2} \times \frac{6\ M\ max}{bh^2}$$

$$= \frac{3Max}{bh^2}$$

$$= \frac{3 \times 84}{bh^2}$$

Allowable stress is 33.3 kpsi

Hence applying Maximum shear stress theory of failure

$$\frac{33.3 \times 1000}{2} = \frac{252}{bh^2}$$

$$\text{or, } (16665) = \frac{252}{bh^2}$$

$$\text{or, } bh^2 = \frac{252}{16665}$$

$$\text{when } b = 0.5\ ; \quad h^2 = \frac{504}{16665}$$

$$h^2 = \frac{504}{16665} \simeq 0.03$$

Therefore, $h = .17$ in. $\cong .2$ inch

When 'b' will be 1.0 inch or 1.5 inches, the value of 'h' will be lower. However for the purpose of convenience and to account for higher factor of safety becuase of unpredictable soil conditions, the value of 'h' for all cases was chosen to be 0.2 inch.

A.3 Design of the Speed Multiplication Unit for the Rotor

Design of Gears

Approach

1. Direction of rotation of rotor-blade is opposite to the direction of movement of the implement.

2. Energy requirement for the rotary-tiller is minimum at a value of $\lambda = 2.4$

3. Approximate horsepower transmitted to the rotor is 1.0 h.p. (>.6 h.p.), since the exact h.p. needed by the machine is not known.

Assumption

Material-AISI 1050 Steel, Drawn at 600^0F

Su = 220,000 psi,

where Su is the ultimate strength of the material (Table A-12) (29)

Theory and Calculations

Factor of safety for the gear

$$ng = k_o k_m n \qquad \text{(Eqn. 11-30)} \quad (29)$$

where ng = special factor of safety for the gear

k_o = an American Gear Manufacturer's Association (AGMA) overload factor

k_m = an AGMA load distribution factor

n = ordinary factor of safety

Referring to Table 11-9 (29) and assuming that the source of power and the driven machinery deliver medium shock to the gear, $k_o = 1.75$.

Assuming that the design is based on static failure

$n = 1.25$

$K_m = 1.6$ (Table 11-10) (29) (assuming that there is less rigid mounting, less accurate gear contact across full face, as far as characteristics of support is concerned; and face width of the gear is 0.2 inch.)

Hence $n_g = 1.75 \times 1.6 \times 1.25$

$= 3.5$

$$\delta_{all} = \frac{S_u}{3.5} = \frac{220{,}000}{3.5} = 62857 \text{ psi}$$

$$h.p. = \frac{W_t \times V}{33{,}000} \qquad \text{(Eqn. 11-13) (29)}$$

where W_t is the load transmitted in lbs.

V = pitch line velocity in feet per minute.

But $V = \frac{\pi d n_1}{12}$

where d = diameter of gear,

n_1 = rpm of the gear.

From modified Lewis Eqn.

$$\delta_{11} = \frac{W_t \times P}{K_v F J} \qquad \text{(Eqn. 11-21) (29)}$$

Where W_t = transmitted load, lbf

$P = \frac{\pi}{p}$

P = diametral pitch, i.e. teeth per inch

$d = \frac{N}{P}$

Where N = No. of teeth

d = pitch diameter in inches.

P = circular pitch in inches.

K_v = velocity factor

F = Face width in inches

J = Geometry factor.

The h.p. transmitted is 1.0.

Referring to the Browning Catalogue, a gear of specification YSS840 was chosen.

Face width of the gear = 1.5 in.

Gear pitch circle diameter 'd' = 5 in.

R.P.M. of the gear $\simeq$ R.P.M. of the walking wheel

Forward speed of machine $\simeq$ 132 ft/min.

Diameter of working wheel = 32/12 = 2.66 ft.

Circumference of walking wheel = π (2.66) = 8.38 ft.

R.P.M. = 132/8.38 = 15.75

Hence N_1 = 15.75

$$V = \pi\, dn/12 = \frac{3.14 \times 5 \times 15.75}{12}$$

$$= 20.60 \text{ ft/min.}$$

$$\text{Hence } 1 = \frac{W_t \times 20.60}{33{,}000}$$

Therefore, $W_t = 1 \times 33{,}000/20.60$

$= 1601.5$ lbs.

Diametral pitch 'P' = 6

No. of teeth 'N' = 30

Referring to Figure 11-20 (29) and assuming that load is applied at the top of the tooth, for 20° pressure angle J = 0.27.

Referring to Figure 11-30 (29)

for V = 20.60,

$$Kv = \frac{50}{50 + \sqrt{V}} = \frac{50}{50 + \sqrt{20.60}}$$

$$= \frac{50}{54.538} = 0.916$$

Therefore, δ actual $= \frac{1601.5 \times 6}{0.916 \times 1.5 \times 0.27}$

= 51, 817.56/2 psi = 25908 psi

which is less than 62857 psi, the allowable stress

Hence the assumptions in selecting the gear are correct.

Design of Chain No. 1 (Ref. Figure 3).

Approach

No. of teeth of pinion = 15

R.P.M. of pinion $= \frac{30 \times 15.75}{15} = 31.5$

Peripheral velocity of rotor = 316.8 ft/min. (132 x 2.4)

Radius of rotor blade = 8/12 ft.

Circumference of the circle described by the rotor blade = 2 x 3.14 x 8/12 = 4.18 ft.

Hence R.P.M. of rotor axle $= \frac{316.8}{4.18} = 75.68$

Therefore, speed multiplication desired by the sprocket chain mechanism is 75.68/31.5 = 2.40.

Assumptions

Type of load = heavy shock

Service conditions = abnormal

Theory and Calculations

Referring to Table 15-14 (29), Load service factor $'k'_s$ for roller chain = 1.4 (choosing 10 hour-day operation). Maximum horsepower to be transmitted = 1.0.

The required h.p. rating = 1 x 1.4 = 1.4

R.P.M. of the sprocket = 15.75 x 2 = 31.50

(on the auxiliary axle number 1)

Referring to the Browning Catalogue (5), for horsepower rating of 1.4, American Standard single strand roller chain, Number 80 or Number 100, may be satisfactory when the lubrication is of Type I (Manual lubricator, oil applied periodically with brush or spout can). Because of space limitation, slower speed and easy commercial availability, a sprocket with 12 teeth, was chosen. This coincides with the view of Shigley (29) that

> 'where space limitations are severe and speed is slow, smaller tooth numbers may be used by sacrificing the life expectancy of the chain.'

From Table 15-13 (29), the 'tooth correction factor' corresponding to 12 tooth sprocket is 0.62. Therefore, (from Browning Catalogue, page 439) the corrected capacity of a Number 80 single strand chain at 31.50 rpm is given by:

Rated hp per strand = 1.23 x 0.62 = 0.762

Similarly the capacity of a Number 100 chain is given by:

Rated hp per strand = 2.30 x 0.62 = 1.426

Number 100 chain is capable of transmitting more than 1.4 horsepower; hence this chain should be used. The extra capacity of

this chain will take care of the unpredictable horsepower requirement.

From Table 15-11 (29), the pitch of Number 100 chain is 1.25. Using a center distance of 9 inches, the required length of single strand chain in pitches is given by:

$$L(p) = \frac{2C}{p} + \frac{N_1 + N_2}{2} + \frac{(N_2 - N_1)^2}{4 \times (c/p)^2} \quad \text{(Eqn. 15.19) (29)}$$

where C = center distance

N_1 = number of teeth in 1st sprocket

N_2 = number of teeth in 2nd sprocket

p = pitch

L(p) = chain length in pitches

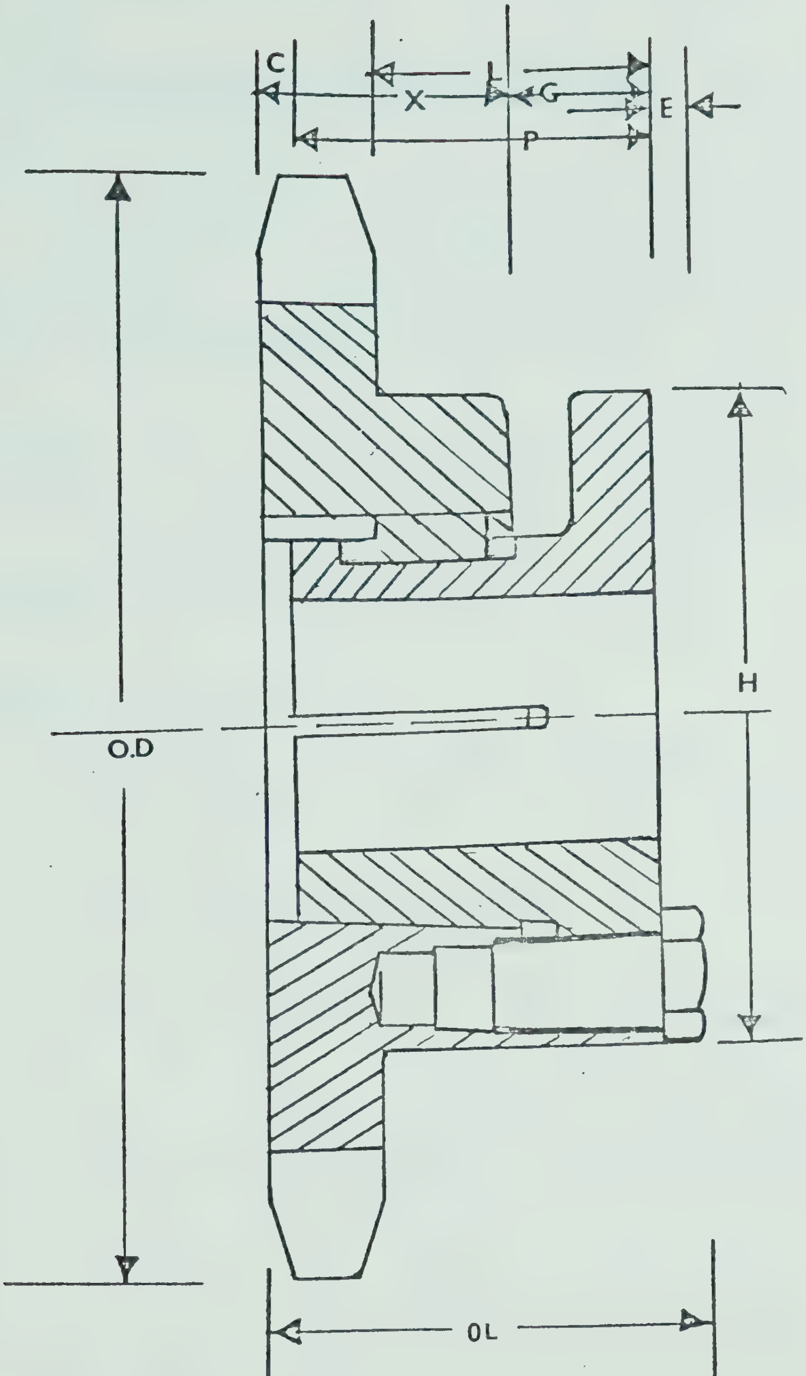
Hence: $$L(p) = \frac{2 \times 9}{1.5} + \frac{12 + 12}{2} + \frac{(12 - 12)^2}{4 \times (9/1.5)^2}$$

$$= 12 + 12$$

$$= 24 \text{ in.}$$

<u>Selection of Sprocket</u>

Referring to Browning Catalogue (5) for a single strand chain of Number 100 the sprocket used is 100 P12, Bushing type being P_1. The specifications of the sprocket and bushing are given in Figure 38.



SPROCKET		BUSHING
OD. = 5.42"	OL. = 2.19"	Bore. = 1.25"
PITCH. = 4.83"	L. = 1.937"	Keyseat = 0.25x0.125"
TYPE. = 4	P. = 1.25"	
T. = .692"	X. = 1.315"	
H. = 3.0"	E. = 0.25"	
G. = 0.625"		

Figure 38. Sprocket No. 100P12; Bushing Type P.

A.4 Design of the Depth Control Screw

Approach

Square and Acme threads are used when power is to be transmitted (29). Because of easy commercial availability an Acme threaded power screw was used.

Assumption

Screw is single threaded. Material of the screw is cold drawn steel (AISI 1010).

Theory and Calculations

Load (F) to be raised = 1500 lbs.

Force needed to raise the load is given by:

$$P = F \left(\frac{\ell + \pi \mu \, dm \, Sec \, \alpha^1}{dm - \mu \ell sec \, \alpha^1}\right)$$

where ℓ = lead, i.e., the distance nut moves parallel to screw axis when the nut is given one turn

μ = coefficient of friction between the thread and the nut

dm = mean diameter of the thread

α^1 = half of the thread angle which is $2\alpha^1$.

In order to raise a load of 1500 lbs.,

Assumed major diameter (d) of screw = 1.0 inch

Pitch (p) = 0.2 inch.

$\alpha^1 = 29°/2 = 14.5°$

Assumed value of μ = 1.0 (Standard)

$dm = d - p/2 = (1.0 - 0.2/2) = 0.9$ inch

$\ell = p$ (for single threaded screw)

Hence $p = 1500 \times [\frac{0.2 + \pi (0.1)(0.9) \text{ Sec } 14.5°}{(0.5) - (0.1)(0.2) \text{ Sec } 14.5°}]$

$= 1500 \times [\frac{0.2 + 3.14(0.1)(0.9) \ (1.05 \ )}{3.14(0.9) \ - (0.1)(0.2)(1.045)}]$

$= 255$ lbs.

Torque required to raise the load is given by:

$T = p \times dm/2$ (Eqn. 7-6)(29)

$= 25 \times 0.9/2$

$= 11$ lbs – inch.

Efficiency of the screw

$e = F\ell/2 \pi T = \frac{1500 \times 0.2}{2 \times 3.14 \times 114}$

$= 0.435$

The force 'F' is transmitted through the screw into the nut. Assuming a factor of safety 'n' of magnitude 4, the maximum allowable stress 'τ max' in the screw material $= \frac{Sy}{2n}$

Where Sy = Yield strength of material

= 44 kps;

Therefore $\tau max = \frac{44}{2 \times 4} = 5.5$ kpsi = 5500 psi

Assuming that load is uniformly distributed over the nut height 'h' and the screw threads would fail by shearing off on the minor diameter, the average screw thread shear stress is given by:

$$\tau = \frac{2F}{\pi \ dr(h)} \qquad \text{(Eqn. 7.7)(29)}$$

Now $\tau = \tau max$

Hence $5500 = \frac{2 \times 1500}{3.14 \times dr(h)}$

But dr = d-p = 1.0 - 0.2 = 0.8 inch

Therefore, $h = \frac{5500 \times 3.14 \times 0.8}{3000}$

$= 4.576$ inches

The threads on the nut would shear off on the major diameter and so the average nut-thread shear stress is given by:

$$\tau n = \frac{2F}{\pi \ dh} \qquad \text{(Eqn. 7.8)(29)}$$

$$= \frac{2 \times 1500}{3.14 \times 1 \times 4.576}$$

$$= 208.0 \text{ psi}$$

Hence if the same material, i.e. AISI 1010 cold drawn steel is used for the nut with a f.s. of 4, the nut will not shear off on the major diameter.

So, material of the nut = AISI 1010 C.D. Steel.

The bearing stress in the thread

$$\sigma = \frac{4pF}{\pi \ h \ (d^2 - dr^2)}$$

$$= \frac{4 \times 0.2 \times 1500}{3.14 \times 4.576 \ (1-0.8^2)}$$

$$= 236.46 \text{ psi.}$$

This provides a large factor of safety which will take care of any bending (the yield strength of the material being 44 kpsi and tensile strength being 53 kpsi).

A.5 Design of the Tension Spring

Approach

The disk tool is forced into the ground by its weight. Consequently standard disk plows are built with heavy frames and wheels (total weight of 400 to 1200 per disk blade). (2)

Assumptions

Total weight needed for two disks = 1500 lbs.

Two tension springs are provided.

The selected spring material is oil tempered wire; diameter of wire = 0.625 inch.

Theory and Calculations

Weight of disks and housing = 50 lbs.

Weight of the frame = 450 lbs.

Hence the weight to be provided by each tension spring on the disk head = $\frac{1500 - 50 - 450}{2}$ = 500 lbs. (This weight is provided by each tension spring when the disks are above the soil surface and not lowered into the ground).

Referring to Figure 39, when the disks are not lowered into the ground, the tension in the spring is maximum and given by:

$$\text{Maximum tension in the spring} = \frac{507.7 \times 29}{5}$$
$$= 2944.0 \text{ lbs.}$$

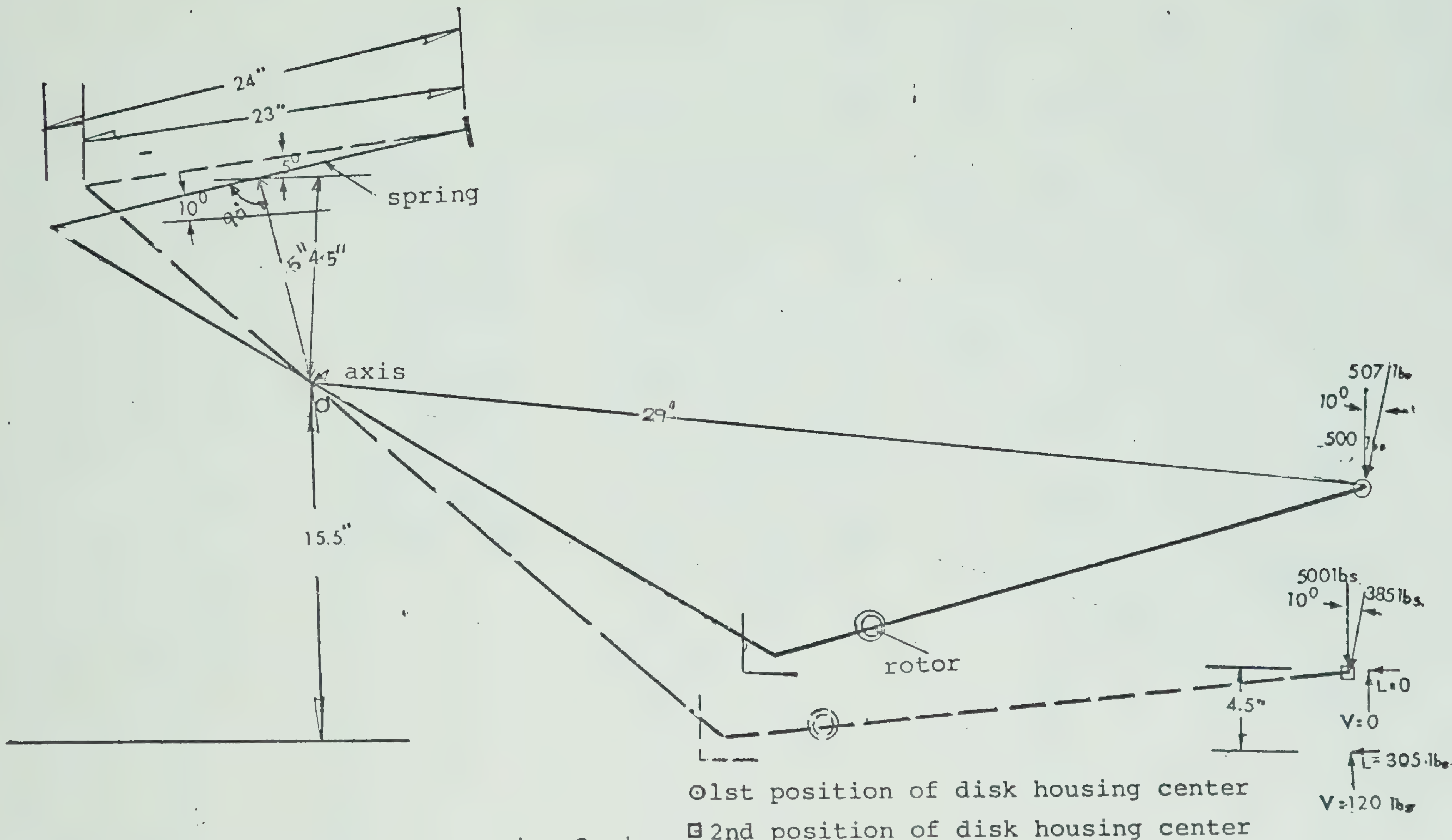


Figure 39. Forces on the Tension Spring.

When the disk is lowered to a depth of 4.5 inches, the tension in the spring is minimum and is given by:

$$\text{Minimum tension} = M_1 - M_2 \text{ Cos } 5^0$$

Where M_1 is the tension needed without draft

M_2 is the tension provided by the draft

$M_2 = \dfrac{305 \times 15.5}{4.5} = 1050$ lbs., $M_1 = \dfrac{385 \times 29}{5} = 2237$ lbs.

$M_2 \text{ Cos } 5^0 = 1050 \times 0.9962$

$= 1040$ lbs.

Therefore, minimum tension in the spring

$= 2237 - 1040 \cong 1197$ lbs.

Deflected length of the spring is equal to 24 inches. When the spring length is 24 inches, the magnitude of the tensile load is equal to 2944 lbs. When the spring length is 23 inches, the magnitude of the tensile load is equal to 1197 lbs.

Reffering to table 8.3 (29)

Su (ultimate tensile strength) = 165 kpsi

Ssy (tensile yield strength) - 0.577 x 0.75 x 165 =

= 71.40 kpsi; (Eqn. 8-10 & 8-11) (29)

$$\text{spring constant 'K'} = \frac{\Delta F}{\Delta Y} = \frac{2944.66 - 1197.94}{24 - 23}$$

$$= 1746.72 \text{ lb /inch.} \qquad \text{(Eqn. 8.2)(29)}$$

$$K = \frac{d^4 G}{8 D^3 N} \qquad \text{(Eqn. 8.8)(29)}$$

Where d = diameter of wire

D = mean spring diameter

G = rigidity modulous (of carbon steel)

$= 11.5 \times 10^6$ psi

N = No. of coils

$$\text{Therefore } ND^3 = \frac{d^4 G}{8K} = \frac{(0.625)^4 \times 11.5 \times 10^6}{8(1746.72)}$$

$$= 123.45$$

Let the solid height of the spring be 'u' inches

$$\text{Hence } K = \frac{2944.66 - 0.0}{24 - u}$$

$$\text{or, } u = 24 - \frac{2944.66}{1746.72}$$

$$= 24 - 1.68$$

$$= 22.32 \text{ inches.}$$

Using short twisted loop at the ends, Number of coils 'N' is given by:

$$N = \frac{22.32 - 1.75}{.625} = 32.912 \simeq 33$$

(Assuming that 1.75 in. is the height of loops)

$$\text{Hence } D^3 = \frac{123.45}{33} = 3.74$$

$$\text{Therefore } D = \sqrt[3]{3.74} = 1.517 \text{ in.}$$

Check

Checking is done to ensure that the stress in the spring while it is in maximum tension is less than the yield strength of the material.

$$\text{Spring Index}, C = D/d = \frac{1.516}{0.625}$$

$$= 2.427$$

Referring to Figure 8.2 (29)

Ks (shear stress multiplication factor)

= 1.20

$$\tau s = \frac{Ks \times 8 \times F \times D}{\pi d^3} \qquad \text{(Eqn. 8.4)(29)}$$

where τs is the stress developed in the spring

$$\tau s = \frac{1.20 \times 8 \times 2944.66 \times 1.517}{3.14 \times (0.625)^3}$$

$$= 55.86 \text{ kpsi } (< 71.50 \text{ kpsi})$$

$$\text{Factor of safety} = \frac{71.40}{55.86} = 1.27$$

Hence the spring with the following dimensions will be suitable :

d = 0.625 inch

D = 1.517 inches

N = 33

A.6 Design of the Compression Spring

Approach

Two compression springs have been provided (Figure 40) in order to allow the coulter blade to move up when

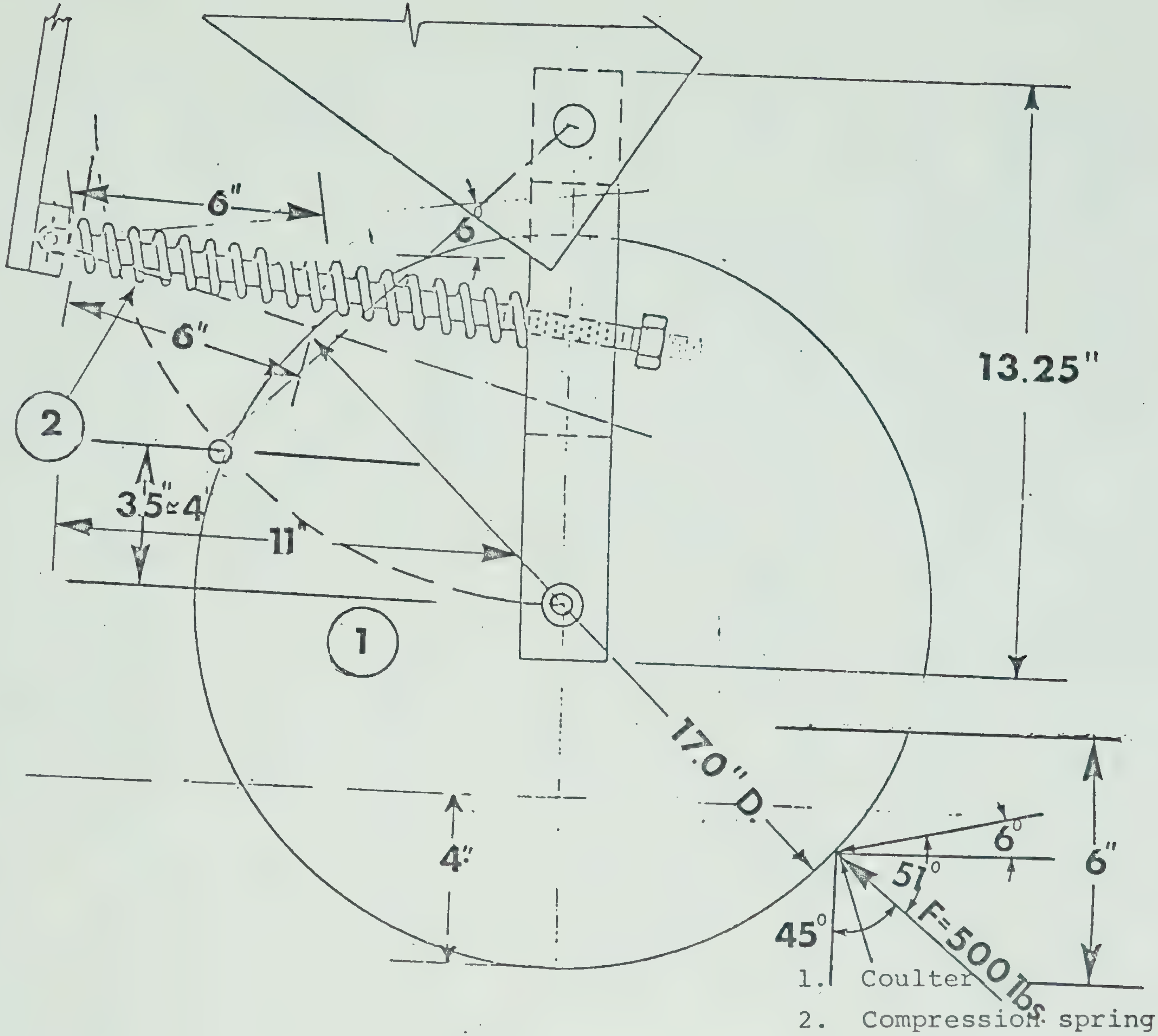


Figure 40. Forces on the Compression Spring.

striking an obstruction.

Assumptions

The coulter is moved up to a height of 4 inches from its bottom position in the ground when striking an obstruction.

Approximate magnitude of the instantaneous force when the coulter strikes an obstruction = 1000 lbs.

Spring material is oil tempered wire with the specification AISI 1065 and wire diameter is 0.312 inch .

Inside diameter of the spring is such that a rod of 1 inch diameter can pass through the spring.

Theory and Calculations

Assuming that the coulter goes to a depth of 6 inches the cross-section area of the furrow = 6 x 0.25 = 1.5 sq.inches Specific Draft requirement = 1.5 x 7 = 10.5 lbs. (Total draft required will be higher because of the weight of the frame on the coulter).

Hence, under normal conditions, each spring should be able to exert an opposite force of 10.5/2 = 5.25 lbs to keep the coulter in its position.

Referring to Figure 40, the desired solid length of the spring (when the coulter moves to the top while striking an obstruction) is 6 inches, when the spring should exert a force of $\left(\frac{1000}{1.5 \times 2}\right)$ = 333.33 lbs. for the return of the spring to the normal position. When the length of the spring is

11 inches, the force exerted by the spring is 5.25 lbs. (The magnitude of the free length of the spring has been assumed according to convenience.)

$$\text{Spring constant } K = \frac{\Delta F}{\Delta Y} = \frac{333.33 - 5.25}{11 - 6}$$

$$= 65.61 \text{ lbs/inch} \qquad \text{(Eqn. 3.2)(29)}$$

$$\text{But } K = \frac{d^4 G}{8D^3 N} \qquad \text{(Eqn. 8.8)(29)}$$

Where d = diameter of wire

D = mean spring diameter

G = Rigidity modulus (of carbon steel) = 11.5×10^6 psi

N = number of coils

$$\text{Therefore } ND^3 = \frac{d^4 G}{8K} = \frac{(0.312)^4 \times (11.5) \times 10^6}{8 \times 65.61}$$

$$= 208.14$$

If the spring has both ends squared and ground, then the total No. of coils = (No. of active coils N) + (2 dead coils). (Section 8.4)(29)

Using 0.321 inch wire, means that fewer than 19 number of coils must be used if the solid height is to be less than 6 inches.

Total number of coils = 19

$$N = 19 - 2 = 17$$

$$D = \sqrt[3]{\frac{208.14}{17}} = \sqrt[3]{12.24} = 2.30 \text{ in.}$$

Inside diameter of the spring = D-d

= 2.305 = 0.312 = 1.993 in.

This satisfies the condition that a rod of 1.0 in. diameter can pass through the spring.

Solid length of the spring 'ℓs' = (N + 2)d

= 19 x 0.312 = 5.928 in.

Spring Index 'C' = $D/d = \frac{2.305}{0.312} = 7.387$

Ks = 1.05 (Ref. Figure 8.2) (29)

Fs = the force required to squeeze the spring solid and is given by:

Fs = (7 - 6.864) x 65.616 + 333.33

= 342.25 lb.

Shear stress in the spring when compressed solid

$= \tau s = Ks \times \frac{8\ Fs\ D}{\pi d^3}$

$= \frac{1.05 \times 8 \times 342.25 \times 2.305}{\pi\ (0.312)^3}$

= 68421.694 psi

= 69.42 kpsi

Allowable yield stress in the spring material

= 183 x .75 x .577

= 79.19 kpsi (> 69.42)

Hence the spring with the following dimensions will hold good.

D = 2.305 in.

d = 0.312 in.

N = 17 (number of active coils)

Total number of coils = 19.

APPENDIX 2

REPLICATE 1 WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatment Days	11	12	13	21	22	23	31	32	33
3rd Day Morning	13	15	18	14	27	22	14	18	23
3rd Day Evening	15	19	19	15	31	27	17	24	25
4th Day Morning	20	19	17	22	28	26	26	32	28
4th Day Evening	22	23	21	30	32	31	32	32	29
5th Day Morning	26	31	32	43	44	44	50	51	36
5th Day Evening	32	33	36	45	50	52	51	54	42
6th Day Morning	51	64	69	50	75	66	58	70	60
6th Day Evening	54	69	71	59	81	72	60	75	70
7th Day Morning	56	69	72	61	82	73	66	76	71
7th Day Evening	58	70	74	63	83	73	69	78	72
8th Day Morning	60	71	74	64	85	74	73	80	79
8th Day Evening	64	72	79	68	90	77	77	83	79
9th Day Morning	68	72	80	69	91	79	82	84	79
9th Day Evening	77	77	81	73	91	83	82	86	79

APPENDIX 3

REPLICATE 1 WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatment Days	11	12	13	21	22	23	31	32	33
3rd Day Morning	10	19	16	14	28	20	14	19	23
3rd Day Evening	13	21	20	15	32	26	16	20	26
4th Day Morning	24	29	29	30	37	28	27	32	27
4th Day Evening	28	32	33	40	40	32	30	33	30
5th Day Morning	31	40	38	43	44	45	36	38	36
5th Day Evening	37	49	47	50	48	50	40	45	44
6th Day Morning	49	60	68	56	75	64	54	70	62
6th Day Evening	55	62	72	58	83	73	62	76	69
7th Day Morning	58	67	77	60	85	74	68	79	72
7th Day Evening	60	70	80	62	87	77	72	81	75
8th Day Morning	64	73	80	65	87	78	75	86	77
8th Day Evening	68	75	80	67	88	79	79	87	80
9th Day Morning	73	76	80	88	90	79	82	97	81
9th Day Evening	76	76	80	71	93	81	83	87	81

APPENDIX 4

REPLICATE 2 WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatments	11	12	13	21	22	23	31	32	33
Days									
3rd Day Morning	12	16	16	13	27	23	14	16	23
3rd Day Evening	14	19	20	14	31	26	16	20	24
4th Day Morning	30	29	24	21	34	29	20	28	30
4th Day Evening	37	35	28	27	38	30	26	34	32
5th Day Morning	44	37	32	29	40	35	29	37	38
5th Day Evening	48	39	40	36	42	39	34	44	39
6th Day Morning	50	62	68	50	75	67	57	70	62
6th Day Evening	56	68	73	58	83	71	60	76	67
7th Day Morning	58	69	74	62	85	72	63	78	69
7th Day Evening	62	70	75	67	87	74	69	80	70
8th Day Morning	66	70	75	69	88	75	77	81	80
8th Day Evening	68	74	76	71	91	79	78	84	80
9th Day Morning	72	76	76	73	91	80	81	84	80
9th Day Evening	73	79	77	74	92	80	82	84	80

APPENDIX 5

REPLICATE 2 WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatments Days	11	12	13	21	22	23	31	32	33
3rd Day Morning	13	16	16	13	25	20	10	18	22
3rd Day Evening	16	21	20	16	33	26	20	22	25
4th Day Morning	20	25	24	22	33	28	24	27	29
4th Day Evening	24	31	27	26	33	29	28	33	33
5th Day Morning	26	37	30	28	36	20	30	38	37
5th Day Evening	29	40	28	32	41	39	34	42	40
6th Day Morning	42	62	68	50	76	68	57	70	63
6th Day Evening	46	70	72	58	77	71	61	74	66
7th Day Morning	47	72	73	61	77	73	68	78	69
7th Day Evening	47	74	76	69	77	74	72	80	71
8th Day Morning	47	76	78	72	77	77	75	81	72
8th Day Evening	48	82	82	72	77	80	79	84	73
9th Day Morning	50	83	84	72	77	81	83	84	79
9th Day Evening	54	84	85	75	78	82	84	85	79

APPENDIX 6

REPLICATE 3 WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatments Days	11	12	13	21	22	23	31	32	33
3rd Day Morning	12	15	13	15	27	23	20	18	24
3rd Day Evening	16	20	21	16	31	26	23	21	27
4th Day Morning	24	27	25	22	32	28	27	28	28
4th Day Evening	29	29	30	28	32	29	30	33	30
5th Day Morning	33	34	32	28	33	30	33	38	37
5th Day Evening	36	29	40	32	37	33	40	44	39
6th Day Morning	50	62	54	50	75	56	40	70	62
6th Day Evening	56	65	69	60	82	56	40	77	68
7th Day Morning	57	67	72	62	87	56	40	79	69
7th Day Evening	58	69	76	63	89	56	41	80	71
8th Day Morning	59	72	76	64	90	56	42	80	75
8th Day Evening	60	74	76	68	91	56	43	80	77
9th Day Morning	65	74	76	70	91	56	43	80	78
9th Day Evening	67	75	76	70	91	56	45	87	79

APPENDIX 7

REPLICATE 3 WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (RAPESEED)

Treatments Days	11	12	13	21	22	23	31	32	33
3rd Day Morning	11	12	18	12	28	22	14	18	20
3rd Day Evening	16	21	20	17	30	28	16	23	27
4th Day Morning	22	27	28	22	30	29	20	24	29
4th Day Evening	27	30	32	27	30	30	24	27	30
5th Day Morning	29	34	35	28	30	32	28	38	32
5th Day Evening	33	39	40	33	39	40	33	39	40
6th Day Morning	52	64	67	50	76	65	50	70	63
6th Day Evening	57	73	71	58	80	73	55	74	69
7th Day Morning	61	73	72	62	84	74	63	76	72
7th Day Evening	63	73	74	68	87	76	67	78	73
8th Day Morning	65	73	75	72	88	77	69	83	75
8th Day Evening	67	74	79	72	90	78	78	84	76
9th Day Morning	70	74	80	72	92	81	82	84	76
9th Day Evening	72	74	80	73	93	82	82	84	79

APPENDIX 8

REPLICATE 1, WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	8	15	15	10	18	18	14	18	13
5th Day Evening	13	19	18	14	25	21	15	21	16
6th Day Morning	18	20	19	15	27	22	17	21	21
6th Day Evening	21	24	20	18	28	23	18	21	24
7th Day Morning	23	27	23	22	32	26	20	22	30
7th Day Evening	27	30	26	24	34	30	23	23	32
8th Day Morning	28	32	28	25	38	34	26	24	36
8th Day Evening	30	38	30	28	40	37	27	28	38
9th Day Morning	32	40	45	37	55	48	35	47	54
9th Day Evening	39	44	55	40	61	54	39	51	58
10th Day Morning	40	44	55	41	63	55	39	53	59
10th Day Evening	42	44	55	41	66	55	39	54	60
11th Day Morning	43	44	55	41	68	55	39	57	61
11th Day Evening	44	44	55	43	70	55	39	59	62
12th Day Morning	44	44	55	43	75	55	39	63	62
12th Day Evening	44	44	55	43	76	55	39	64	62
13th Day Morning	44	44	55	43	76	55	39	65	62
13th Day Evening	44	44	55	43	76	55	39	65	62
14th Day Morning	44	44	55	43	76	55	39	65	62
14th Day Evening	44	46	55	44	76	55	39	69	63

APPENDIX 9

REPLICATE 1, WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	8	17	16	10	17	18	13	19	20
5th Day Evening	12	19	20	16	23	20	17	22	21
6th Day Morning	16	21	23	18	28	24	19	25	26
6th Day Evening	19	24	25	20	30	25	21	29	29
7th Day Morning	20	26	28	24	33	29	25	34	31
7th Day Evening	23	28	30	25	35	31	27	36	33
8th Day Morning	24	29	32	27	37	33	28	37	34
8th Day Evening	25	31	35	29	37	36	28	39	37
9th Day Morning	32	39	47	37	58	48	34	50	54
9th Day Evening	36	41	51	39	61	58	38	51	56
10th Day Morning	36	41	52	39	68	54	38	54	56
10th Day Evening	36	41	54	40	70	56	39	56	56
11th Day Morning	36	41	55	42	73	57	39	58	56
11th Day Evening	36	42	57	42	75	57	39	61	56
12th Day Morning	36	42	58	42	77	57	39	64	56
12th Day Evening	36	42	58	42	78	57	39	68	56
13th Day Morning	36	42	58	42	78	57	39	72	56
13th Day Evening	36	42	58	42	79	57	39	73	56
14th Day Morning	36	42	58	42	79	57	39	73	56
14th Day Evening	36	42	59	43	79	58	41	77	57

APPENDIX 10

REPLICATE 2, WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	9	14	16	12	21	18	14	18	17
5th Day Evening	13	18	18	15	25	21	17	22	21
6th Day Morning	16	20	23	17	29	23	19	24	25
6th Day Evening	18	23	26	19	31	25	24	27	28
7th Day Morning	20	26	31	21	34	26	26	29	33
7th Day Evening	23	27	34	22	35	29	27	31	36
8th Day Morning	24	29	38	24	39	31	28	34	39
8th Day Evening	26	31	40	26	44	34	31	38	40
9th Day Morning	35	40	46	36	60	50	39	49	54
9th Day Evening	37	43	50	40	61	54	41	52	59
10th Day Morning	38	44	52	42	64	55	42	56	59
10th Day Evening	39	44	54	42	69	57	42	59	59
11th Day Morning	39	44	57	42	71	58	42	63	59
11th Day Evening	39	44	58	42	73	58	42	66	59
12th Day Morning	40	44	59	42	75	58	42	68	59
12th Day Evening	41	44	60	42	77	58	42	73	59
13th Day Morning	42	44	61	42	78	58	42	75	59
13th Day Evening	42	44	61	42	79	58	42	77	59
14th Day Morning	42	44	61	42	79	58	42	78	59
14th Day Evening	42	44	62	42	81	59	42	79	60

APPENDIX 11

REPLICATE 2, WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	9	12	13	13	20	12	14	17	16
5th Day Evening	13	18	17	16	28	16	16	21	19
6th Day Morning	19	20	20	19	30	18	21	23	23
6th Day Evening	21	24	24	22	32	22	24	24	26
7th Day Morning	23	26	27	28	35	24	26	27	28
7th Day Evening	27	27	29	30	39	26	28	29	30
8th Day Morning	28	29	31	31	40	28	30	34	32
8th Day Evening	30	31	33	32	42	30	31	36	37
9th Day Morning	34	44	48	37	58	52	38	48	56
9th Day Evening	38	46	52	41	62	54	40	52	58
10th Day Morning	40	46	53	42	67	55	41	63	59
10th Day Evening	41	48	57	42	69	57	41	68	60
11th Day Morning	41	48	59	42	73	57	41	69	60
11th Day Evening	41	48	60	42	74	57	41	73	60
12th Day Morning	41	48	61	42	75	57	41	75	60
12th Day Evening	41	48	61	42	79	57	41	76	60
13th Day Morning	41	48	61	42	80	58	41	76	60
13th Day Evening	41	48	61	42	81	58	41	77	60
14th Day Morning	41	48	61	42	81	58	41	78	60
14th Day Evening	41	49	62	43	82	59	42	79	60

APPENDIX 12

REPLICATE 3, WITHOUT COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	8	15	16	10	20	16	15	18	14
5th Day Evening	12	20	19	14	28	20	16	22	18
6th Day Morning	18	24	23	18	30	28	18	24	24
6th Day Evening	21	28	25	23	33	33	20	26	28
7th Day Morning	23	30	27	24	37	34	24	29	29
7th Day Evening	27	33	29	26	39	35	28	30	32
8th Day Morning	28	34	30	28	40	37	30	34	34
8th Day Evening	30	35	33	30	43	39	33	36	37
9th Day Morning	33	42	50	36	61	50	37	49	52
9th Day Evening	37	44	55	38	64	52	41	51	56
10th Day Morning	38	45	57	40	66	53	41	58	56
10th Day Evening	39	45	60	40	68	55	41	60	56
11th Day Morning	40	45	60	40	70	57	41	63	58
11th Day Evening	40	45	60	40	72	58	41	64	58
12th Day Morning	40	45	60	40	73	60	41	67	59
12th Day Evening	40	45	60	40	73	60	41	69	59
13th Day Morning	40	45	60	40	73	60	41	71	59
13th Day Evening	40	45	60	40	73	60	41	73	59
14th Day Morning	40	45	60	40	73	60	41	75	59
14th Day Evening	40	46	64	42	74	60	42	76	59

APPENDIX 13

REPLICATE 3, WITH COULTER

NUMBER OF FULLY EMERGED SEEDLINGS (WHEAT)

Treatments Days	11	12	13	21	22	23	31	32	33
5th Day Morning	10	15	14	10	20	16	15	10	14
5th Day Evening	12	18	19	14	25	20	18	14	18
6th Day Morning	14	20	21	18	29	24	20	18	22
6th Day Evening	17	23	23	20	31	24	22	20	24
7th Day Morning	19	26	24	21	33	29	23	23	26
7th Day Evening	23	28	26	23	37	32	24	27	29
8th Day Morning	24	30	27	24	39	33	26	29	31
8th Day Evening	26	31	29	24	41	34	27	31	33
9th Day Morning	33	39	50	34	60	48	39	54	54
9th Day Evening	37	42	55	38	68	52	41	56	54
10th Day Morning	39	43	58	38	70	52	41	63	58
10th Day Evening	40	43	60	39	72	53	41	66	58
11th Day Morning	41	43	61	40	73	54	41	68	58
11th Day Evening	42	43	62	41	74	56	41	72	58
12th Day Morning	42	43	62	42	75	58	41	73	58
12th Day Evening	42	43	62	42	77	60	41	77	58
13th Day Morning	42	43	62	42	78	60	41	78	58
13th Day Evening	42	43	62	42	79	60	41	78	58
14th Day Morning	42	43	62	42	79	60	41	79	58
14th Day Evening	43	43	64	43	81	60	41	79	58

APPENDIX 14

PERCENTAGE OF SEEDLING EMERGENCE (WHEAT) AT THE END OF THE 5th DAY

Treatment	Replicate	% of emergence without coulter	% of emergence with coulter
11	1	10.5	10.0
11	2	11.0	11.0
11	3	10.0	11.0
21	1	12.0	13.0
21	2	13.5	14.5
21	3	12.0	12.0
31	1	14.5	15.0
31	2	16.5	15.0
31	3	15.5	16.5
12	1	17.0	18.0
12	2	16.0	15.0
12	3	17.5	16.5
22	1	21.5	20.0
22	2	23.0	24.0
22	3	24.0	22.5
32	1	19.5	20.5
32	2	20.0	19.0
32	3	20.0	19.0
13	1	16.5	18.0
13	2	17.0	15.0
13	3	17.5	17.0
23	1	19.5	19.0
23	2	19.5	19.0
23	3	18.0	18.0
33	1	17.0	21.0
33	2	19.5	17.5
33	3	16.0	16.0

APPENDIX 15

PERCENTAGE OF SEEDLING EMERGENCE (WHEAT)
AT THE END OF THE 9th DAY

Treatment	Replicate	% of emergence without coulter	% of emergence with coulter
11	1	35.5	34.0
11	2	36.0	36.0
11	3	35.0	36.0
21	1	38.5	38.0
21	2	38.0	39.0
21	3	37.0	36.0
31	1	37.0	36.0
31	2	40.0	39.0
31	3	39.0	40.0
12	1	42.0	40.0
12	2	41.5	45.0
12	3	43.0	40.5
22	1	58.0	59.5
22	2	60.5	60.0
22	3	62.5	64.0
32	1	49.0	50.5
32	2	50.5	50.0
32	3	50.0	55.0
13	1	50.0	49.0
13	2	48.0	50.0
13	3	52.5	50.5
23	1	51.0	50.5
23	2	52.0	53.0
23	3	51.0	50.0
33	1	56.0	55.0
33	2	55.5	57.0
33	3	57.0	54.0

APPENDIX 16

PERCENTAGE OF SEEDLING EMERGENCE (WHEAT)
AT THE END OF THE 14th DAY

Treatment	Replicate	% of emergence without coulter	% of emergence with coulter
11	1	45.5	36.0
11	2	42.0	41.0
11	3	40.0	42.5
21	1	43.5	42.5
21	2	42.0	42.5
21	3	41.0	42.5
31	1	39.0	40.0
31	2	42.0	41.5
31	3	41.5	41.0
12	1	45.0	42.0
12	2	44.0	48.5
12	3	45.5	43.0
22	1	76.0	79.0
22	2	80.0	81.5
22	3	73.5	80.0
32	1	67.0	75.0
32	2	78.5	78.5
32	3	75.5	79.0
13	1	55.5	58.5
13	2	61.5	61.5
13	3	62.0	63.0
23	1	55.0	57.5
23	2	58.5	58.5
23	3	60.0	60.0
33	1	62.5	56.5
33	2	59.5	60.0
33	3	59.0	58.0

APPENDIX 17

PERCENTAGE OF SEEDLING EMERGENCE (RAPESEED)

THE END OF 3rd DAY

Treatment	Replicate	% Emergence	
		Without Coulter	With Coulter
11	1	14.0	13.0
11	2	13.0	14.5
11	3	14.0	13.5
21	1	14.5	14.5
21	2	13.5	14.5
21	3	15.5	14.5
31	1	15.5	15.0
31	2	15.0	15.0
31	3	21.5	15.0
12	1	17.0	20.0
12	2	17.5	18.5
12	3	17.5	16.5
22	1	29.0	30.0
22	2	29.0	29.0
22	3	29.0	29.0
32	1	20.0	19.5
32	2	18.0	20.0
32	3	19.5	20.5
13	1	18.5	18.0
13	2	18.0	18.0
13	3	17.0	19.0
23	1	24.5	23.0
23	2	24.5	23.0
23	3	24.5	25.0
33	1	24.0	24.5
33	2	23.5	23.5
33	3	25.5	23.5

APPENDIX 18

PERCENTAGE OF SEEDLING EMERGENCE (RAPESEED)

THE END OF 6th DAY

Treatment	Replicate	% Emergence	
		Without Coulter	With Coulter
11	1	52.5	52.0
11	2	53.0	44.0
11	3	53.0	54.5
21	1	54.5	54.0
21	2	54.0	54.0
21	3	55.0	54.0
31	1	59.0	57.0
31	2	58.5	59.0
31	3	40.0	52.5
12	1	66.5	61.0
12	2	65.0	66.0
12	3	63.5	68.5
22	1	78.0	79.0
22	2	79.0	76.5
22	3	78.0	78.0
32	1	72.5	73.0
32	2	73.0	72.0
32	3	73.5	72.0
13	1	70.0	70.0
13	2	70.5	70.0
13	3	67.0	69.0
23	1	69.0	68.5
23	2	69.0	69.5
23	3	56.0	69.0
33	1	65.0	65.5
33	2	64.5	64.5
33	3	65.0	66.0

APPENDIX 19

PERCENTAGE OF SEEDLING EMERGENCE (RAPESEED)

THE END OF 9th DAY

Treatment	Replicate	% Emergence	
		Without Coulter	With Coulter
11*	1	72.5	74.5
11	2	72.5	52.0
11	3	66.0	71.0
21	1	71.0	69.5
21	2	73.5	72.5
21	3	70.0	72.5
31	1	82.0	82.5
31	2	81.5	83.5
31	3	44.0	82.0
12	1	74.5	76.0
12	2	77.5	77.5
12	3	74.5	74.0
22	1	91.0	91.5
22	2	91.5	92.0
22	3	91.0	92.5
32	1	85.0	87.0
32	2	84.0	84.5
32	3	83.5	84.0
13	1	80.5	80.0
13	2	76.5	80.0
13	3	76.0	80.0
23	1	80.5	80.5
23	2	80.0	81.5
23	3	56.0	81.5
33	1	79.0	81.0
33	2	80.0	79.0
33	3	78.5	77.5

*11: width 1, height 1
11: width 1: 0.5"; width 2: 1.0"; width 3: 1.5"
height 1: 4.5"; height 2: 6.75"; height 3: 9.0".

APPENDIX 20

PULL (RAPESEED PLOT)

Treatment	Replicate	Pull (lbs)	
		without coulter	with coulter
11	1	450	350
11	2	400	350
11	3	400	300
21	1	450	350
21	2	400	350
21	3	450	350
31	1	500	400
31	2	550	400
31	3	450	350
12	1	500	400
12	2	400	300
12	3	400	350
22	1	500	350
22	2	400	400
22	3	400	350
32	1	650	300
32	2	500	425
32	3	450	400
13	1	600	400
13	2	550	350
13	3	450	350
23	1	650	450
23	2	550	300
23	3	500	400
33	1	650	500
33	2	400	400
33	3	550	450

APPENDIX 21

PULL (WHEAT PLOT)

Treatment	Replicate	Pull (lbs) without coulter	Pull (lbs) with coulter
11	1	550	400
11	2	550	550
11	3	450	400
21	1	550	500
21	2	650	400
21	3	600	400
31	1	600	400
31	2	750	400
31	3	550	450
12	1	450	350
12	2	400	350
12	3	450	350
22	1	400	350
22	2	400	350
22	3	550	300
32	1	450	300
32	2	400	300
32	3	400	300
13	1	450	400
13	2	550	400
13	3	500	600
23	1	500	400
23	2	600	400
23	3	600	450
33	1	450	350
33	2	500	300
33	3	450	300